NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1206

IN-LINE AIRCRAFT-ENGINE BEARING LOADS

III - MAIN-BEARING LOADS

By Milton C. Shaw and E. Fred Macks

Aircraft Engine Research Laboratory
Cleveland, Ohio

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SUMMARY

Dimensional analysis has been applied to the computation of the main bearing of a V-type in-line aircraft engine. Charts are presented that give the maximum and mean bearing loads for the center, intermediate, and end main bearings at all values of engine speed to 5000 rpm and at all values of indicated mean effective pressure to 500 pounds per square inch. Two crankshafts have been considered, one designed for higher operating speed than the other. The individual effects of changes in compression ratio and in the ratio of connecting-rod length to crank throw upon the main-bearing loads are presented.

Optimum combinations of engine speed and indicated mean effective pressure were found to exist for which the mean and maximum main-bearing loads are minimums for a given power but such combinations do not always lie in a practicable operating region. Polar diagrams of bearing loads are presented to show the extent of shock load and the range of stress imposed upon the main bearings for several operating conditions.

INTRODUCTION

A general method of determining aircraft-engine bearing loads at any combination of engine speed and indicated mean effective pressure has been given in reference 1. This method has been applied to determine the loads acting on the crankpin and blade bearings of a V-type engine in references 1 and 2, respectively. The following equation, relating the significant engine operating variables to the load acting on any of the principal bearings of an aircraft engine, is obtained by dimensional analysis:

$$W = L_S{}^2\, p\,\Omega\left(\frac{M_1N^2}{L_Sp}, \frac{M_cN^2}{L_Sp}, \frac{D}{L_S}, \frac{L_R}{L_S}, \frac{p_m}{p}, \theta, r, \gamma\right) \qquad (1)$$

where

W bearing load, pounds

L_S stroke, inches

p indicated mean effective pressure, pounds per square inch

Ω some function

M_i reciprocating mass per crankpin, slugs

N engine speed, rpm

M_c rotating mass per crankpin, slugs

D diameter of bore, inches

L_R length of connecting rod, inches

p_m manifold pressure, pounds per square inch absolute

θ crank angle, degrees

r compression ratio

γ angle between cylinder center lines, degrees

If the indicated mean effective pressure is assumed to be proportional to the manifold pressure, equation (1) simplifies to the following expression for a given engine:

$$W = p\,\Omega'\left(\frac{N^2}{p}\ \theta\right) \tag{2}$$

This equation establishes the fact that, if W/p is plotted against N^2/p at a constant value of crank angle, a smooth curve will be obtained. Equations (1) and (2) are applicable to the principal bearings of both in-line and radial engines.

In the present report, mean and maximum main-bearing loads are discussed under a wide range of operating conditions. Two crankshafts of different design are investigated (fig. 1): a 6-counterweight crankshaft, which will be referred to as "crankshaft A," and a 12-counterweight crankshaft, designed for higher operating speeds, which will be referred to as "crankshaft B."

Dimensions of a production V-type engine are used to illustrate the application of the generalized method to the computation of main-bearing loads. Before equation (2) can be applied and before the computations can be generalized, the main-bearing loads for a number of representative engine operating conditions must be computed to obtain values of W/p. These computations are made in the usual manner (reference 3).

CONVENTIONAL COMPUTATION OF MAIN-BEARING LOADS WITH CRANKSHAFT A

The symbols, conventions, engine dimensions, power conditions, and method of analysis used in this report are the same as those of reference 1. Specifications for the V-type engine investigated herein are given in the appendix. A zero crank angle ($\theta = 0$) refers to the top-center position of cylinder 1L at the beginning of the expansion stroke. A schematic diagram of a 12-cylinder V-type engine mechanism is given in figure 2 and the crankshaft arrangement is shown diagrammatically in figure 3.

The resultant main-bearing loads were obtained by the vector addition of the crankpin loads and the centrifugal forces due to the unbalanced weights of the crankpin and the crankcheeks. Each crankpin load is assumed to be divided equally between its adjacent main bearings. The unbalanced weight of the crankshaft between transverse crankpin midsections is assumed to act upon the included main bearing.

The method employed to obtain the unbalanced crankshaft weights is given in reference 4. Crankshaft A is symmetrical about the center main bearing and the following values of unbalanced force were found to act in the direction of the reference crankpins:

Main bearing	1	2	3	4	5	6	7
Reference crankpin	1	2	3	-	4	5	6
Force due to unbalanced crankshaft weight, lb $\times 10^{-6}N^2$	336	336	336	0	336	336	336

Center main bearing. - The resultant load acting on the center main bearing (bearing 4 in fig. 3) at any particular crank angle is obtained by the vector addition of one-half the adjacent crankpin loads. The centrifugal force is zero inasmuch as counterbalancing is complete for this particular bearing. Figure 4 illustrates the method of vector addition for the center main bearing at a crank-angle value of 200°.

Polar diagrams of loads acting on the center main bearing with respect to the engine axis are shown in figure 5 in terms of crank-angle degrees.

Intermediate main bearings. - The resultant load acting on the intermediate main bearings (bearings 2, 3, 5, and 6 in fig. 3) at any particular crank angle is obtained by the vector addition of the unbalanced centrifugal force acting along the crank-throw center line and one-half of each of the adjacent crankpin loads. Figure 6 illustrates the method of vector addition for main bearing 2 at a crank-angle value of 20°.

Because of the phase relation of the piston displacement and the symmetry of the crankshaft, the polar diagram for main bearing 3 may be obtained from the diagram for main bearing 2 by adding 240° to each indicated value of crank angle. Thus the configurations of these two diagrams are the same but the values of crank angle obtained at each point are different. Similarly, the diagrams for main bearings 5 and 6 may be obtained by adding 600° and 360°, respectively, to the values appearing on the diagram for main bearing 2.

Polar diagrams, with respect to the engine axis, of the loads acting on the intermediate main bearings are shown in figure 7 in terms of crank-angle degrees.

End main bearings. - The resultant load acting on either end main bearing (bearing 1 or 7 in fig. 3) at any particular crank angle is obtained by the vector addition of the unbalanced centrifugal force acting along the crank-throw center line and one-half of the single adjacent crankpin load. Figure 8 illustrates the method of vector addition for main bearing 1 at a crank-angle value of 20°. The polar diagram for main bearing 7 may be obtained from the diagram for main bearing 1 by adding 360° to each indicated value of crank angle.

Polar diagrams of the loads acting on the end main bearings, with respect to the engine axis, are shown in figure 9 in terms of crank-angle degrees.

APPLICATION OF THE DIMENSIONAL-ANALYSIS METHOD

Generalized load charts for maximum loads on center main bearing with crankshaft A. - The resultant forces on the center main bearing shown in figure 5 can be generalized by equation (2). If W/p is plotted against N^2/p for each of the six power conditions considered and at constant values of crank angle, it is found that maximum values of W/p occur at crank-angle values of approximately 20° or 320°. These curves are shown in figure 10; additional points were computed in order to extend the curves beyond the region covered by the six power conditions. The solid portion of each curve corresponds very closely to the maximum value of W/p over the particular range of N^2/p concerned. All the plots of W/p against N^2/p are portions of hyperbolic-type curves. The solid portions of both curves of interest lie sufficiently far from their respective vertices to be considered linear.

A convenient chart (fig. 11) for determining maximum center main-bearing loads is obtained from the curves of figure 10. Curves of constant indicated horsepower have been included for convenience. The line OA represents an optimum combination of engine speed and indicated mean effective pressure corresponding to the lowest value of the maximum load on the center main bearing at a given power level. It can be seen from figure 11 that this optimum combination does not fall in a practicable operating region.

Generalized load charts for mean loads on center main bearing with crankshaft A. - The mean load acting on the center main bearing is determined from a rectangular plot of load against crank angle using a planimeter to obtain the average height of this curve. The results of the dimensional treatment were again utilized to generalize the mean-load analysis. In figure 12, $\overline{W}/p$ is shown plotted against N^2/p, where $\overline{W}$ is the mean bearing load.

A useful chart (fig. 13) for determining mean center main-bearing loads is obtained from the curves of figure 12. Constant indicated horsepower curves have been included for convenience.

Rubbing factor. - The "rubbing factor," as defined in reference 4, may be obtained from the value of the mean load from figure 12 for any combination of engine speed and indicated mean effective pressure and the equation

$$RF = 2.30 \times 10^{-3}(N\,\overline{W}) \qquad (3)$$

where RF is the rubbing factor in foot-pounds per square inch-second. The coefficient in this equation is based upon an effective bearing length of 1.90 inches. Although the rubbing factor is not generally considered a good criterion for the severity of bearing operating conditions, it is given for what it may be worth.

<u>Generalized load charts for intermediate and end main bearings with crankshaft A.</u> - Maximum and mean bearing-load charts have been prepared for the intermediate and the end main bearings (figs. 14 to 17) using the procedure employed in obtaining the charts for the center main-bearing loads. Optimum maximum-load lines shown for four compression ratios were included in figures 15 and 17 to permit a comparison with each other and with the mean-load curves and will be discussed later in this report.

The rubbing factors for the intermediate and the end main bearings may be calculated from the following equation in which $\overline{W}$ is the appropriate mean bearing load obtained from figures 15 to 17:

$$RF = 3.97 \times 10^{-3}(N\,\overline{W}) \tag{4}$$

The coefficient in this equation is based upon an effective bearing length of 1.10 inches. (All intermediate and end main bearings have the same effective length.)

CHARTS FOR CENTER, INTERMEDIATE, AND END MAIN BEARINGS WITH CRANKSHAFT B

Crankshaft B is also symmetrical about the center main bearing and has the following values of unbalanced force:

Main bearing	1	2	3	4	5	6	7
Force due to unbalanced crankshaft weight, lb $\times 10^{-6}$ N^2	140	140	140	280	140	140	140
Reference crankpin	1	2	3	3,4	4	5	6
Angle, in degrees, from the reference crank-throw center line to the line connecting the center of gravity of the unbalanced force and the center of rotation measured in the direction of crankshaft rotation	180	240	240	180	120	120	180

Polar diagrams with respect to the engine axis for the take-off condition are given in figures 18 to 20; maximum and mean load charts are presented in figures 21 to 26 for a compression ratio of 6.65.

EFFECT OF ENGINE DIMENSIONS UPON MAIN-BEARING LOADS

Attempts to make the load charts applicable to any in-line engine have not been entirely successful. No simple method has been found by which a change in the magnitude of the reciprocating and rotating weights may be taken into consideration by the application of dimensional analysis. The main-bearing loads will not be influenced by a change in the ratio of connecting-rod length to crank throw in the range from 3 to 4, inasmuch as changes within this range were found in reference 1 to have no measurable effect upon crankpin bearing loads.

The compression ratio affects the shape of the indicator diagram and hence the gas force developed in the engine cylinder. The influence of compression ratio upon gas force during the exhaust stroke, the intake stroke, and most of the compression stroke is quite small. The compression ratio has a considerable effect upon the gas force, however, during that portion of the expansion stroke when the piston is near the top-center position.

The compression ratio will influence the mean main-bearing load very little, inasmuch as the compression ratio significantly affects the gas force only during a small portion of the cycle and part of this effect is compensatory. The mean-load diagrams shown in figures 13, 15, 17, 22, 24, and 26 are applicable for all values of compression ratio from 5.50 to 8.50.

The maximum load acting on the center main bearing is not affected by compression ratio because of the particular phasing of events in the adjacent cylinders and therefore figure 11 is also applicable for all compression ratios from 5.50 to 8.50.

The curves of figures 14 and 16 showing maximum loads for the intermediate and the end main bearings with crankshaft A for a compression ratio of 6.65 are supplemented by figures 27 and 28 for compression ratios of 5.50, 7.50, and 8.50. Similar charts are not given for crankshaft B but the approximate magnitude of the effect of compression ratio may be obtained from a comparison of figures 14, 16, 23, 25, 27, and 28. The locations of the curves OA for optimum combinations of speed and indicated mean effective pressure, for the intermediate and the end main bearings, change with compression ratio.

The OA curves for the four compression ratios here considered have been included in figures 15 and 17. It is evident that, as the compression ratio is increased, the optimum indicated mean effective pressure for a given engine speed is less.

DISCUSSION

The maximum and mean loads acting on the crankpin, blade, and main bearings under a wide range of engine operating conditions for the V-type engine herein considered are summarized in tables I and II. For the bearing metals used none of the unit loads appear to be excessive. Although the maximum load acting on the blade bearing is considerably higher than corresponding loads on the other bearings, the high rate of load fluctuation enables this bearing to operate satisfactorily. It is evident that the maximum load acting on the center and the end main bearings is significantly reduced when crankshaft B is substituted for crankshaft A. The load acting on the intermediate main bearings, however, is greater with crankshaft B than with crankshaft A except at very high speeds. The mean load acting on all of the main bearings is lower with crankshaft B than with crankshaft A.

From the charts of references 1 and 2 and this report, it is evident that the crankpin, blade, and main-bearing loads with crankshaft A are not significantly increased at values of indicated mean effective pressure up to 200 pounds per square inch at 3000 rpm. The end and center main-bearing loads with crankshaft B do not vary greatly with indicated mean effective pressure at a constant engine speed; however, the loads on the intermediate main bearings increase rapidly with indicated mean effective pressure. Because it is inadvisable to operate with closed throttle setting in a dive owing to increased oil pumping under such conditions, both crankshafts may be operated at values of indicated mean effective pressure up to 200 pounds per square inch, unless difficulty is experienced with the intermediate bearings of crankshaft B, in which case the indicated mean effective pressure should be held to an absolute minimum.

In this report and in references 1 and 2 an optimum combination of engine speed and indicated mean effective pressure refers to one for which the mean or maximum bearing load is a minimum. Such a combination is optimum only with regard to the mechanics of the problem, giving rise to a minimum unit load. From the consideration of hydrodynamic lubrication the load capacity of a bearing, as measured by the quantity $\frac{\text{(viscosity)(engine speed)}}{\text{(unit bearing load)}}$ is a more important criterion. Thus, if the viscosity is considered constant, the load capacity of the bearing will vary as the ratio of engine speed to unit bearing load. Minimum load and hydrodynamic optimums are given in table III for the principal bearings of the V-type engine herein considered. The two series of maximum optimums are, in general, the same; whereas the corresponding values of mean optimums always differ. Although the values of optimums given in table III do not always lie in the practicable region of operation, this table may be used to ascertain whether a shift of the operating conditions of the engine in the direction of the optimum will be of benefit to a bearing giving trouble.

The foregoing analysis of main-bearing loads has been carried out in the conventional manner in which deflection of the crankshaft and the crankcase is neglected. Such deflections are known to exist and might significantly alter the distribution of load among the several main bearings. An analytical determination of main-bearing loads including crankshaft and crankcase deflections is not practical and experimental investigations such as those described in references 4 to 6 should be made. So little work of this nature has been published that a rational determination of the distribution of in-line engine main-bearing loads cannot be made.

CONCLUSIONS

From a series of computations using the dimensional-analysis method of analyzing the main-bearing loads of a V-type in-line aircraft engine the following conclusions are drawn:

For V-type engines:

1. Minimum load-optimum combinations of engine speed and indicated mean effective pressure exist for which the mean and maximum main-bearing loads are minimums for a given power output.

2. Hydrodynamic-optimum combinations of engine speed and indicated mean effective pressure exist for which the ratio of engine speed to mean or maximum unit bearing load is a maximum for a given power output. The hydrodynamic and minimum-load optimums are generally the same for maximum loads, whereas the corresponding values for mean loads differ.

3. The maximum and mean center main-bearing loads are independent of compression ratio, whereas the maximum intermediate and end main-bearing loads vary directly with compression ratio for a given engine speed and indicated mean effective pressure. The intermediate and end main-bearing mean loads are independent of compression ratio.

4. The ratio of connecting-rod length to crank throw does not appreciably influence the mean or maximum main-bearing loads.

For the production V-type engine herein considered:

1. The combinations of engine speed and indicated mean effective pressure corresponding to optimum values of both mean and maximum main-bearing loads do not always lie in a practicable operating region.

2. With regard to crankpin blade and main bearings, both the low-speed and the high-speed crankshafts may be operated at values of indicated mean effective pressure up to 200 pounds per square inch in a dive unless difficulty is experienced with the intermediate bearings of the high-speed shaft, in which case the indicated mean effective pressure should be held to an absolute minimum.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio. June 21, 1946.

APPENDIX - SPECIFICATIONS OF A PRODUCTION

12-CYLINDER, V-TYPE ENGINE

Number of cylinders	12
Arrangement of cylinders	Two blocks at an angle of 60°
Method of numbering cylinders from antipropeller end, both blocks	1, 2, 3, 4, 5, 6
Firing order	1L, 2R, 5L, 4R, 3L, 1R, 6L, 5R, 2L, 3R, 4L, 6R
Direction of crankshaft rotation, viewing antipropeller end	Counterclockwise
Bore, in.	5.50
Stroke, in.	6.00
Piston area, sq in.	23.75
Engine speed at take-off, rpm	3000
imep at take-off, lb/sq in.	242
bmep at take-off, lb/sq in.	186
Manifold pressure at take-off, in. Hg absolute	52
Assumed mechanical efficiency at take-off, percent	77
Compression ratio	6.65
Fork-rod length, in.	10.00
Blade-rod length, in.	10.00
Ratio of connecting-rod length to crank throw	3.33
Spark advance, deg B.T.C.:	
Intake	28
Exhaust	34
Valve timing:	
Intake valve open, deg B.T.C.	48
Intake valve closed, deg A.B.C.	62
Exhaust valve open, deg B.B.C.	76
Exhaust valve closed, deg B.T.C.	26
Blade bearing:	
Diameter, in.	3.70
Effective length, in.	0.93
Projected area, sq in.	3.44
Crankpin bearing:	
Diameter, in.	3.00
Effective length, in.	1.94
Projected area, sq in.	5.81
Center main bearing:	
Diameter, in.	3.75
Effective length, in.	1.90
Projected area, sq in.	7.12

Intermediate and end main bearings:
- Diameter, in. 3.75
- Effective length, in. 1.10
- Projected area, sq in. 4.13

Reciprocating and rotating weights:
- Weight of piston assembly, lb 5.31
- Average weight of upper end of blade or fork rod, lb 1.41
- Reciprocating weight per cylinder, lb 6.72
- Weight of crankpin bearing, lb 2.21
- Weight of lower end of fork rod, lb 3.10
- Weight of lower end of blade rod, lb 2.60
- Rotating weight per crankpin, lb 7.91

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TABLE I - REPRESENTATIVE VALUES OF CRANKPIN, BLADE, AND MAIN MAXIMUM UNIT BEARING LOADS[1] FROM THE CHARTS OF REFERENCES 1 AND 2 AND THIS REPORT

	Power condition								
	1	2	3	4	5	6	7	8	9
Engine speed, rpm	3000	3300	3600	3000	3000	3000	3000	3000	3000
imep, lb/sq in.	242	242	242	182	303	363	242	242	242
Indicated horsepower	1570	1720	1880	1170	1960	2350	1570	1570	1570
Compression ratio	6.65	6.65	6.65	6.65	6.65	6.65	5.50	7.50	8.50
Crankpin bearing	2670	3120	3610	2540	3300	4320	2640	2770	3320
Blade bearing	4700	4250	3640	2920	6480	8230	3160	5450	5960
Center main bearing with crankshaft A	2010	2360	2760	1950	2060	2140	2010	2010	2010
Intermediate main bearing with crankshaft A	3010	3380	3870	2730	3500	4140	2920	3260	3530
End main bearing with crankshaft A	2600	3070	3550	2480	2740	2800	2620	2620	2600
Center main bearing with crankshaft B	1660	1910	2260	1580	1740	1820	1660	1660	1660
Intermediate main bearing with crankshaft B	3330	3425	3500	2460	3860	4650	----	----	----
End main bearing with crankshaft B	1830	1800	2070	1453	2610	3290	----	----	----

[1]The unit loads in lb/sq in. are based upon the following values of bearing area:

Bearing	Projected area (sq in.)
Crankpin	5.81
Blade	3.44
Center main	7.12
Intermediate and end main	4.13

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TABLE II - REPRESENTATIVE VALUES OF CRANKPIN, BLADE, AND MAIN MEAN UNIT BEARING LOADS[1] FROM THE CHARTS OF REFERENCES 1 AND 2 AND THIS REPORT

	Power condition								
	1	2	3	4	5	6	7	8	9
Engine speed, rpm	3000	3300	3600	3000	3000	3000	3000	3000	3000
imep, lb/sq in.	242	242	242	182	303	363	242	242	242
Indicated horsepower	1570	1720	1880	1170	1960	2350	1570	1570	1570
Compression ratio	6.65	6.65	6.65	6.65	6.65	6.65	5.50	7.50	8.50
Crankpin bearing	2030	2320	2660	1910	2170	2300	2030	2030	2030
Blade bearing	1730	1940	2200	1560	1920	2060	1730	1730	1730
Center main bearing with crankshaft A	1350	1660	2020	1410	1450	1600	1350	1350	1350
Intermediate main bearing with crankshaft A	2100	2300	2680	1890	2330	2580	2100	2100	2100
End main bearing with crankshaft A	2050	2420	3110	2000	2090	2120	2050	2050	2050
Center main bearing with crankshaft B	1100	1330	1580	1090	1210	1350	1100	1100	1100
Intermediate main bearing with crankshaft B	1640	1770	1920	1380	1910	2410	1640	1640	1640
End main bearing with crankshaft B	1240	1400	1570	1120	1380	1510	1240	1240	1240

[1]The unit loads in lb/sq in. are based upon the following values of bearing area:

Bearing	Projected area (sq in.)
Crankpin	5.81
Blade	3.44
Center main	7.12
Intermediate and end main	4.13

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TABLE III - TABLE OF OPTIMUM OPERATING CONDITIONS FOR THE PRINCIPAL BEARINGS OF THE V-TYPE ENGINE CONSIDERED IN REFERENCES 1 AND 2 AND THIS REPORT

Bearing	Compression ratio	N^2/p			
		Minimum-load optimum		Hydrodynamic optimum	
		For maximum load	For mean load	For maximum load	For mean load
Crankpin	5.50	28,000	6,500	28,000	13,500
	6.65	34,000	6,500	34,000	13,500
	7.50	39,000	6,500	39,000	13,500
	8.50	43,500	6,500	43,500	13,500
Blade	5.50	47,500	12,500	47,500	25,700
	6.65	54,000	12,500	54,000	25,700
	7.50	60,000	12,500	60,000	25,700
	8.50	67,000	12,500	67,000	25,700
Center main with crankshaft A	5.50-8.50	16,500	22,500	14,500	27,500
Intermediate main with crankshaft A	5.50	22,500	20,000	22,500	30,000
	6.65	26,000	20,000	26,000	30,000
	7.50	27,000	20,000	27,000	30,000
	8.50	29,000	20,000	29,000	30,000
End main with crankshaft A	5.50	18,000	15,000	18,000	17,500
	6.65	23,000	15,000	23,000	17,500
	7.50	25,500	15,000	25,500	17,500
	8.50	29,000	15,000	29,000	17,500
Center main with crankshaft B	6.65	15,000	27,500	15,000	32,500
Intermediate main with crankshaft B	6.65	33,500	35,000	33,500	50,000
End main with crankshaft B	6.65	42,500	20,000	42,500	25,000

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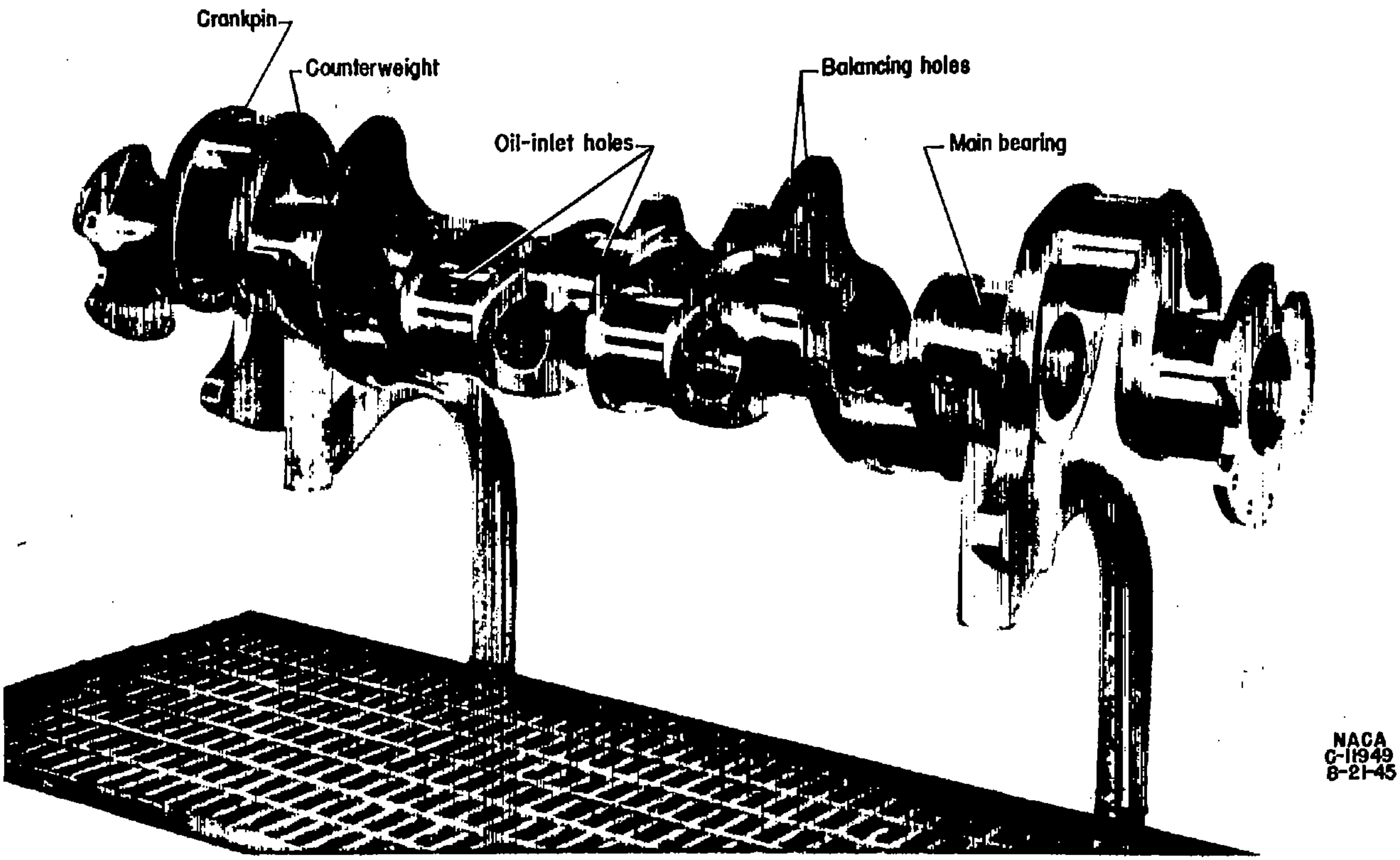
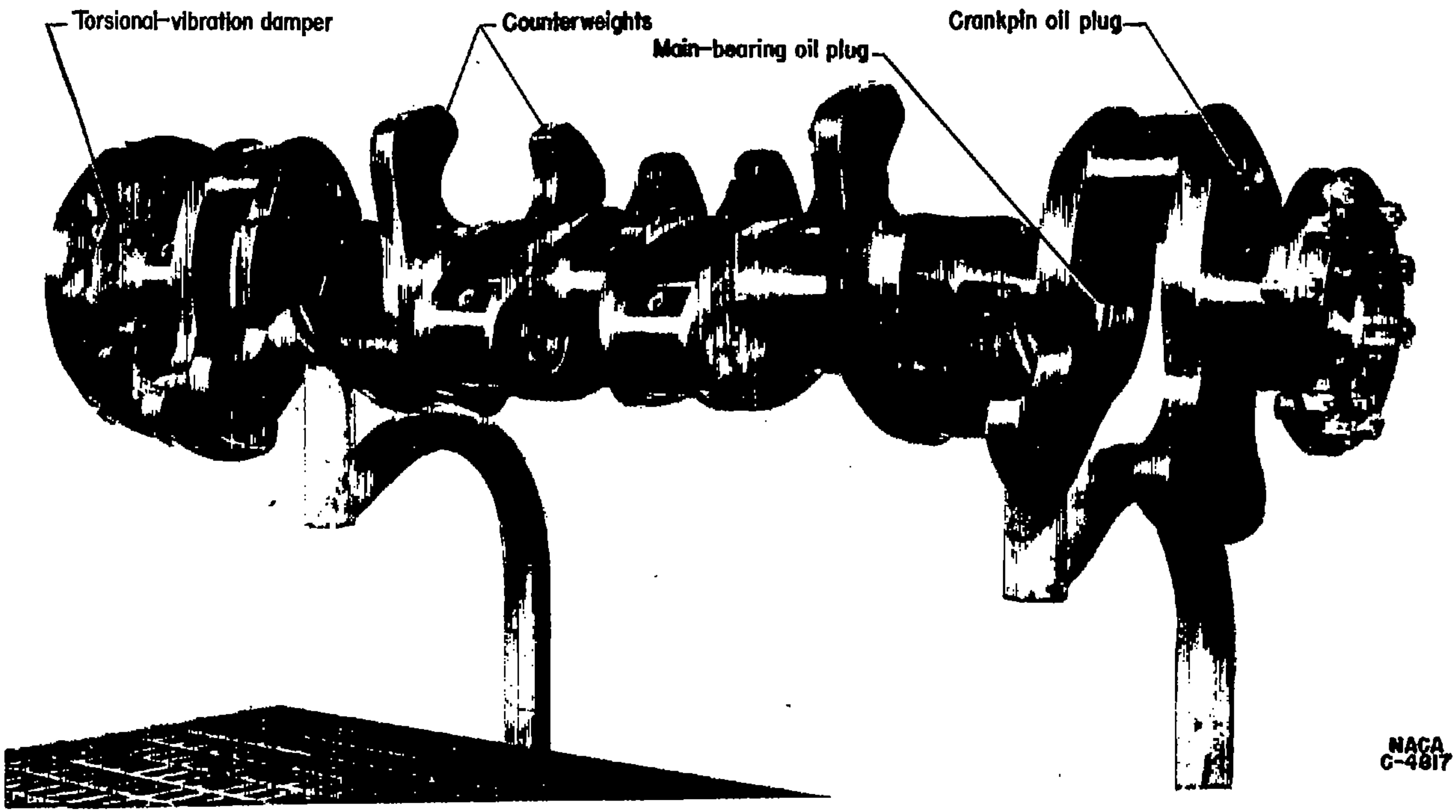


(a) Crankshaft A.

Figure 1. - Photographs showing two types of crankshafts for a 12-cylinder, V-type engine.



(b) Crankshaft B.

Figure 1. - Concluded.

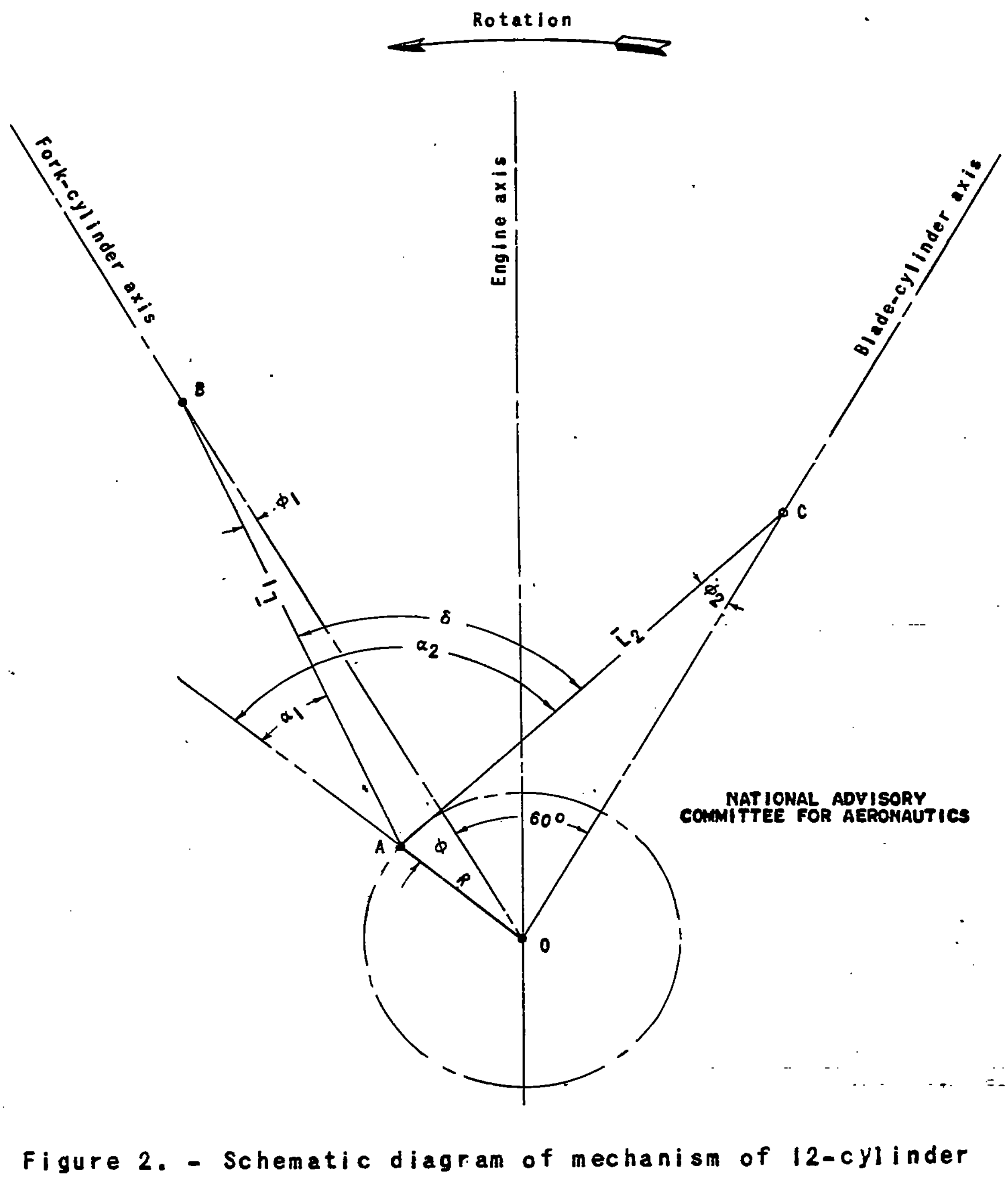


Figure 2. - Schematic diagram of mechanism of 12-cylinder V-type engine.

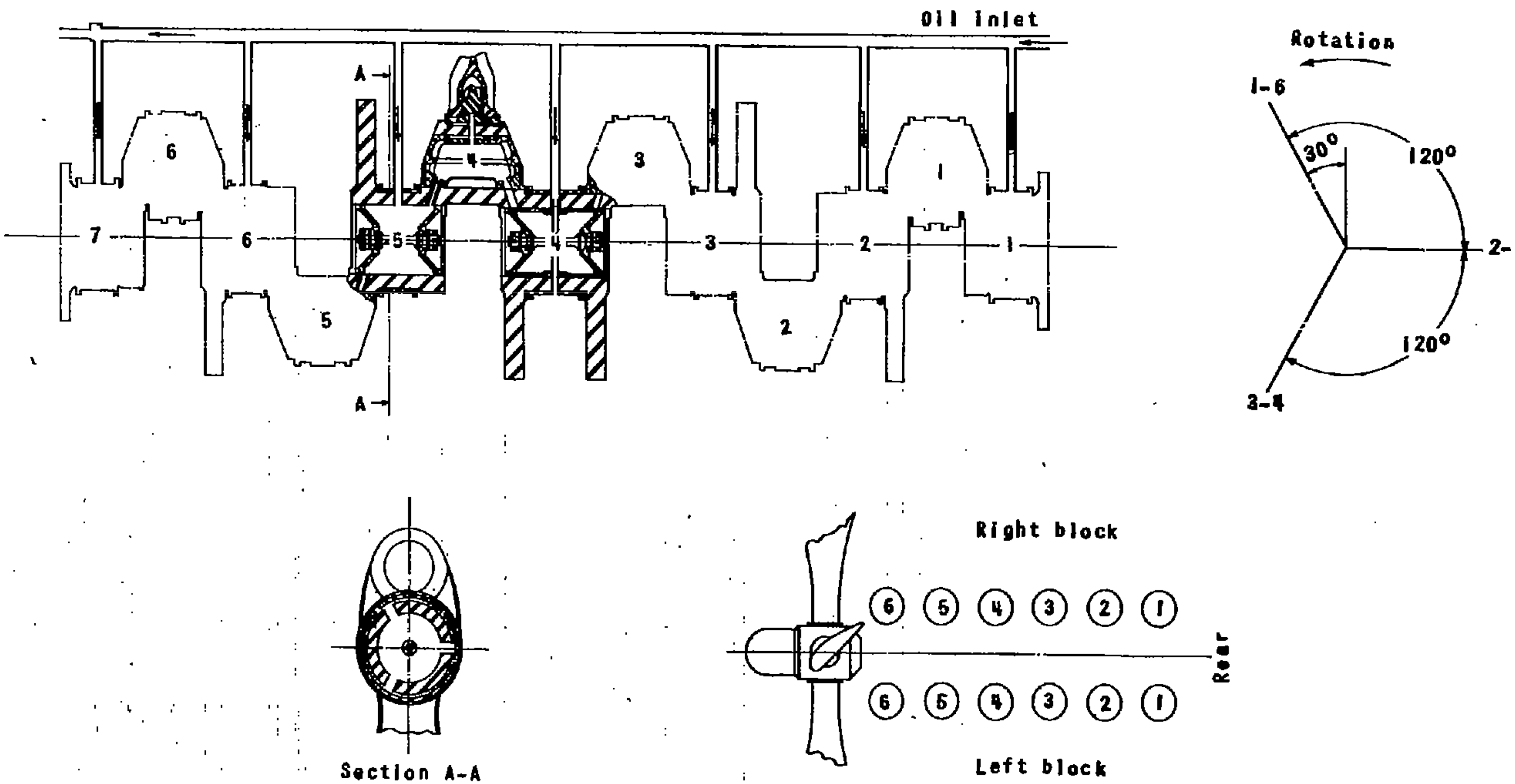


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Figure 3. - Diagrammatic view of crankshaft arrangement of 12-cylinder v-type engine.

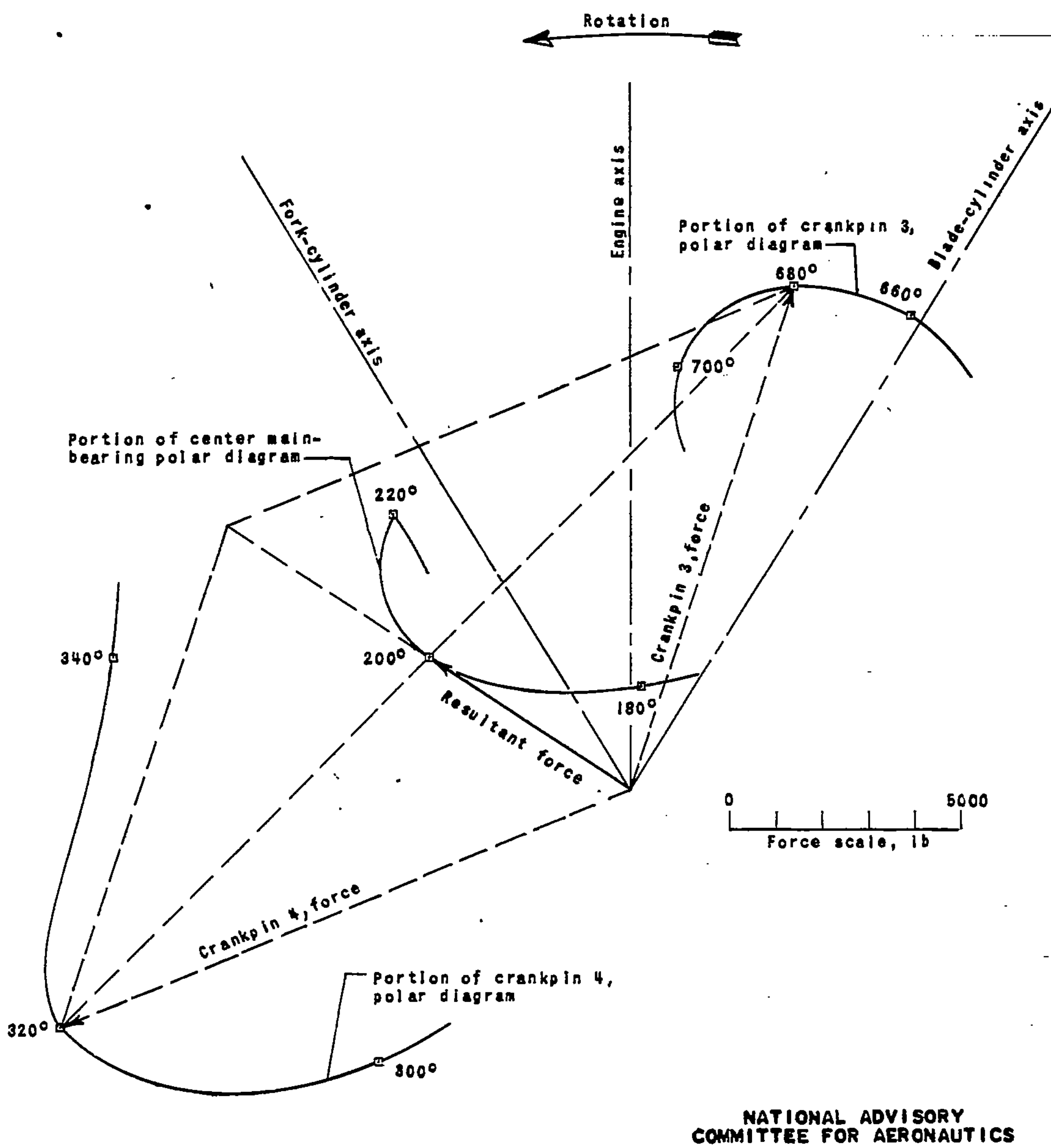
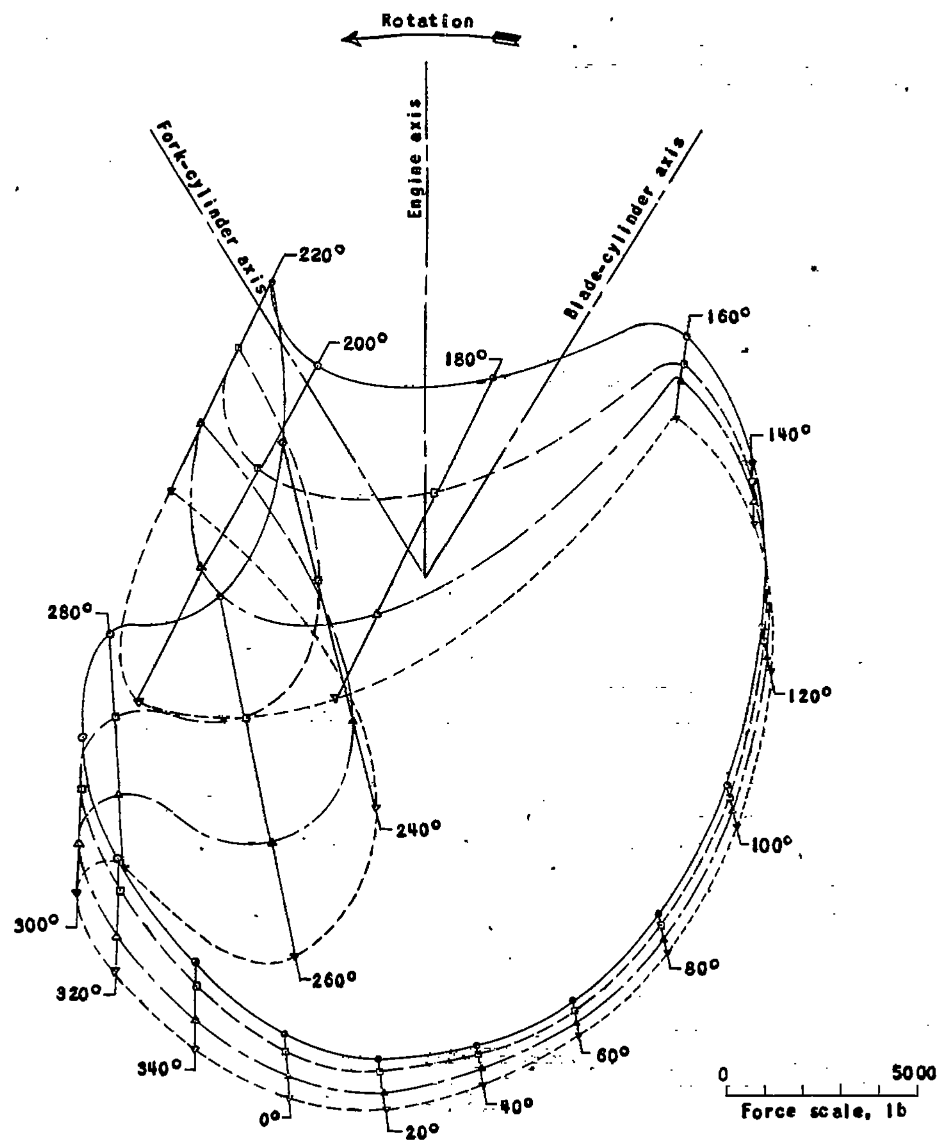


Figure 4. - Representative vector addition necessary to obtain the polar diagram of the load acting upon the center main bearing of a V-type engine with crankshaft A.



	imep	Maximum force (lb)	Mean force (lb)
	182	13,880	10,040
	242	14,310	9,610
	303	14,670	10,320
	363	15,240	11,390

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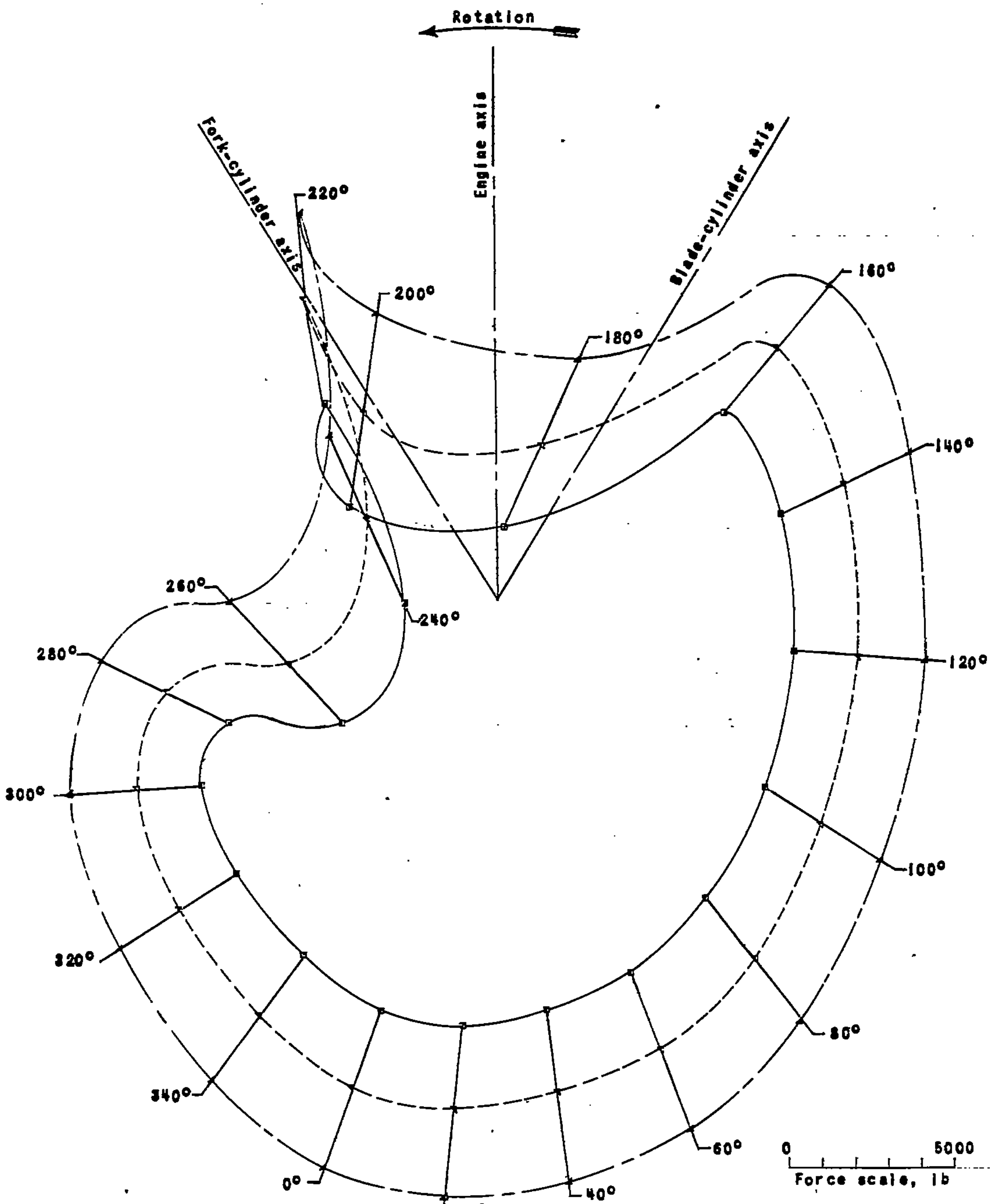
Indicated crank-angle degrees are for main bearing 2; add 240°, 600°, and 360° in the direction of rotation for main bearings 3, 5, and 6 respectively.

(a) Engine speed, 3000 rpm; indicated mean effective pressures, 182, 242, 303, and 363 pounds per square inch.

Figure 5. - Polar diagrams showing the magnitude of the resultant force on the center main bearing of V-type engine with crankshaft A and its direction with respect to the engine axis at different power conditions.



rpm	Maximum force (lb)	Mean force (lb)
3000	14,310	9,610
3300	16,800	11,820
3600	19,650	14,380

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Indicated crank-angle degrees are for main bearing 2; add 240°, 600°, and 360° in the direction of rotation for main bearings 3, 5, and 6, respectively.

(b) Indicated mean effective pressure, 242 pounds per square inch; engine speeds, 3000, 3300, and 3600 rpm.

Figure 5. - Concluded.

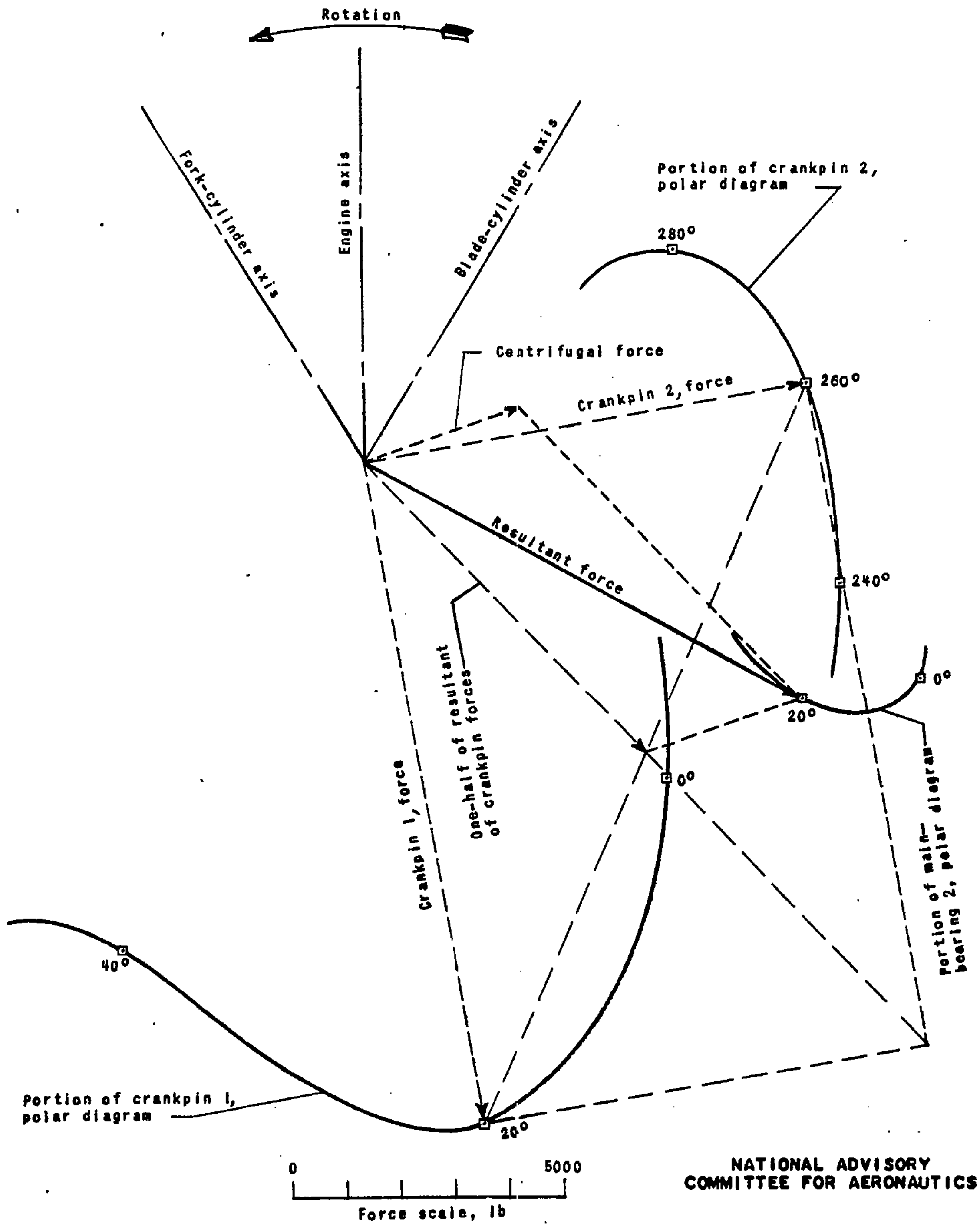

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Figure 6. - Representative vector addition necessary to obtain the polar diagram of the load acting upon intermediate main bearing 2 of a V-type engine with crankshaft A.

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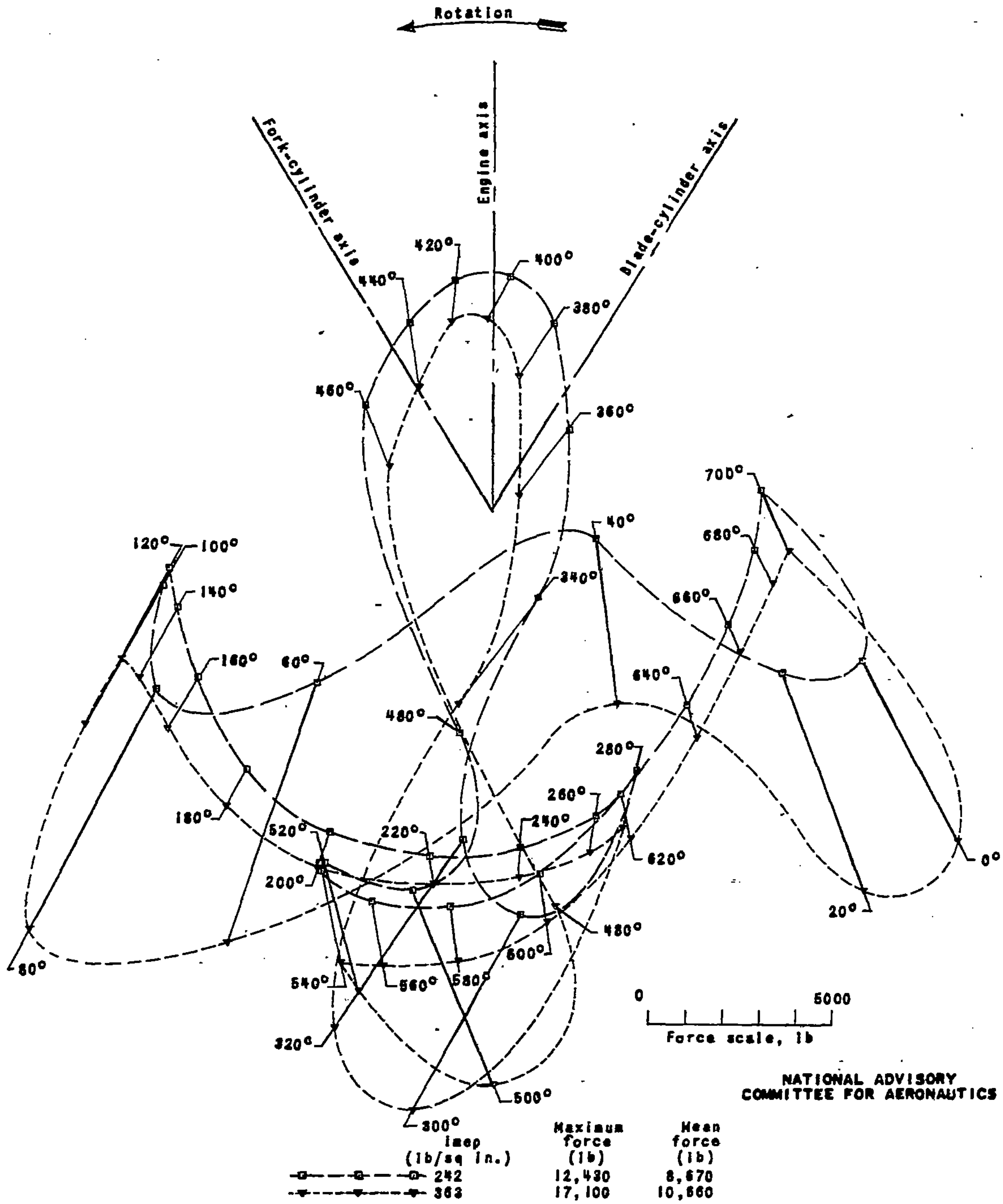
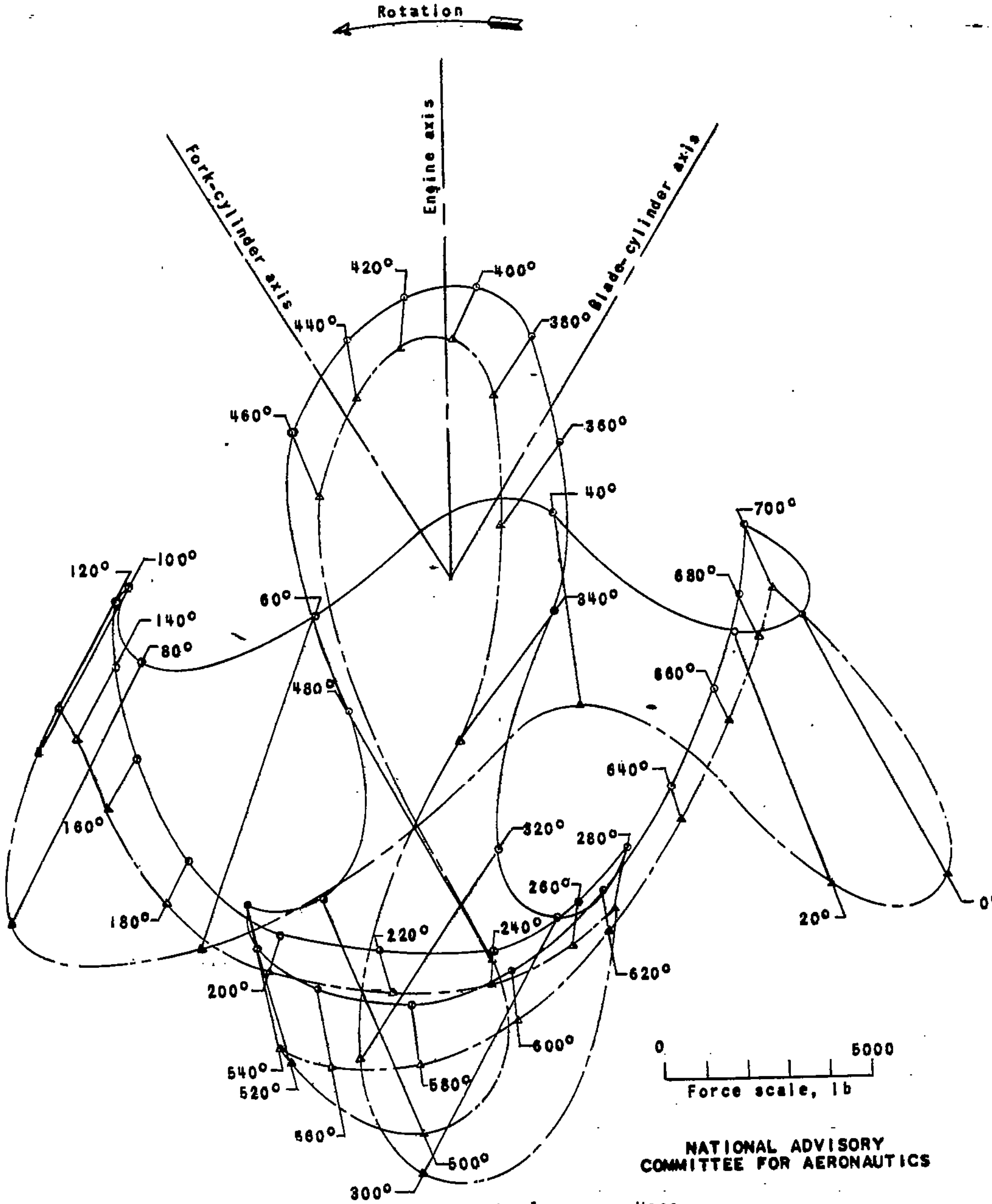


Indicated crank-angle degrees are for main bearing 2; add 240°, 600°, and 360° in the direction of rotation for main bearings 3, 5, and 6 respectively.

(a) Engine speed, 3000 rpm; indicated mean effective pressures, 242 and 363 pounds per square inch.

Figure 7. - Polar diagrams showing the magnitude of the resultant force on intermediate main bearings 2, 3, 5, and 6 of a V-type engine with crankshaft A and its direction with respect to the engine axis at different power conditions.



imep (lb/sq in.)	Maximum force (lb)	Mean force (lb)
182	11,280	7,810
303	14,370	9,620

Indicated crank-angle degrees are for main bearing 2; add 240°, 600°, and 360° in the direction of rotation for main bearings 3, 5, and 6, respectively.

(b) Engine speed, 3000 rpm; indicated mean effective pressures, 182 and 303 pounds per square inch.

Figure 7. - Continued.

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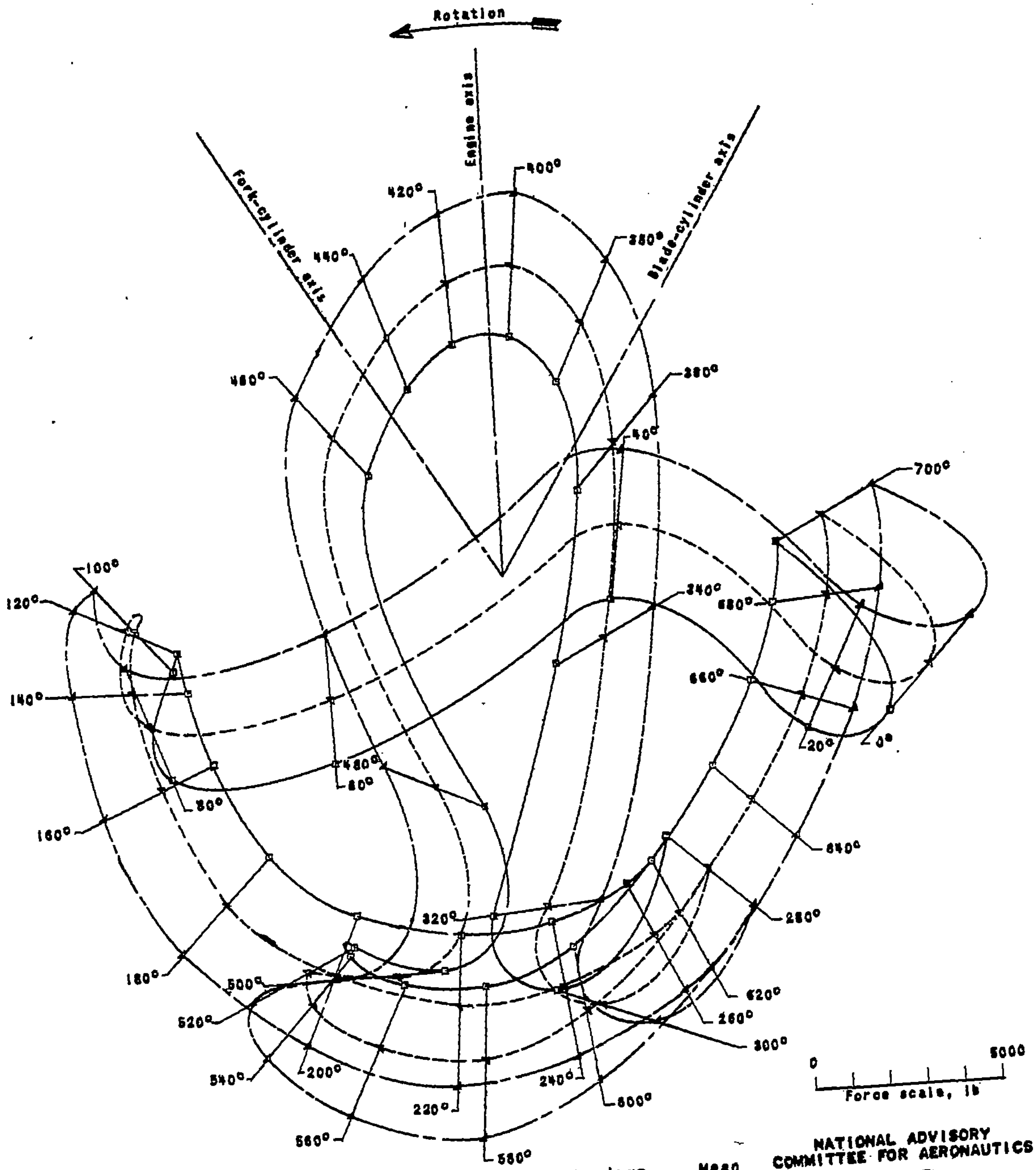

Engine speed (rpm)	Maximum force (lb)	Mean force (lb)
3000	12,430	8,670
3300	13,960	9,500
3600	15,980	11,070

Indicated crank-angle degrees are for main bearing 2; add 240°, 600°, and 360° in the direction of rotation for main bearings 3, 5, and 6, respectively.

(c) Engine speeds, 3000, 3300, and 3600 rpm; indicated mean effective pressure, 242 pounds per square inch.

Figure 7. - Concluded.

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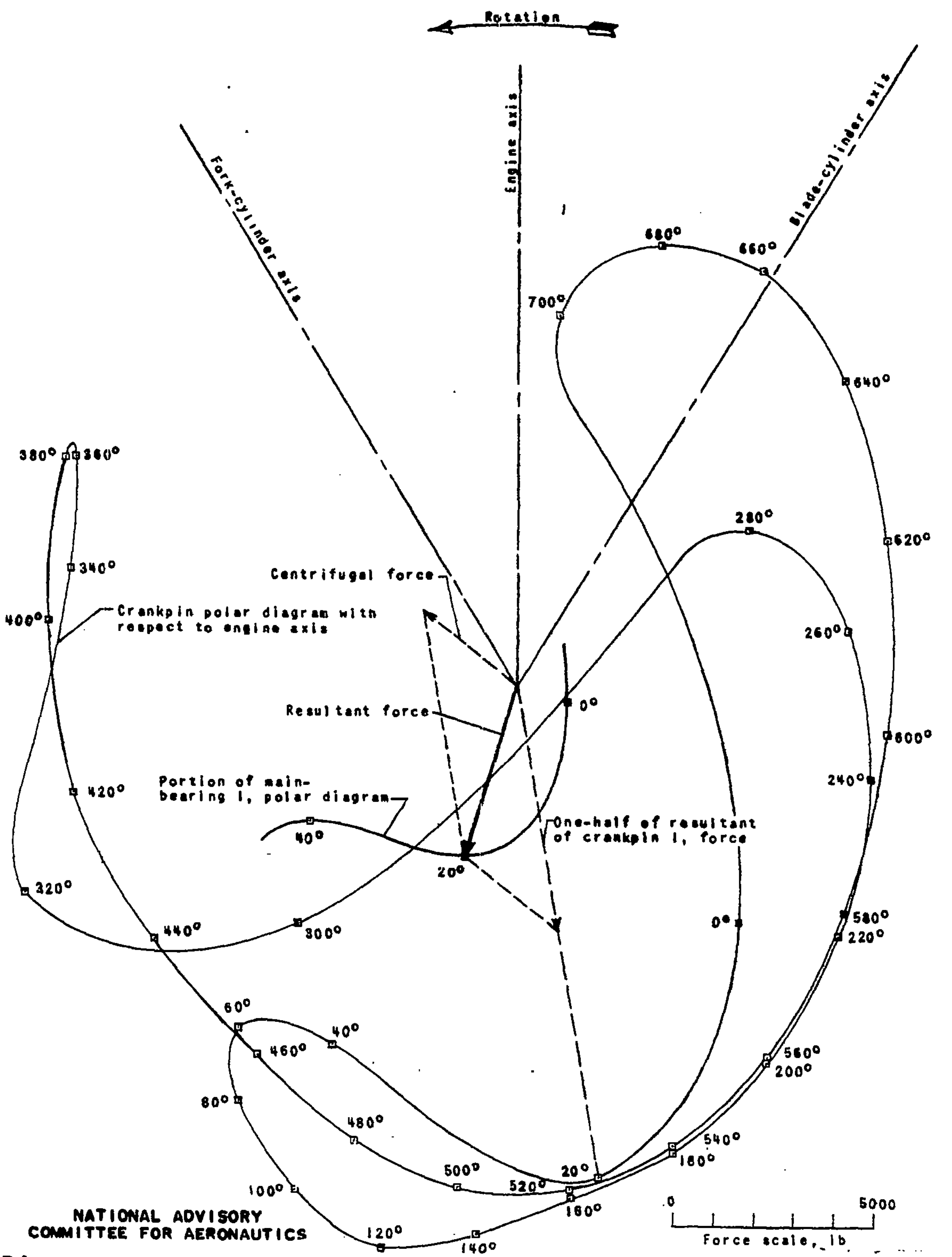

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Figure 8. - Representative vector addition necessary to obtain the polar diagram of the load acting upon end main bearing I of a V-type engine with crankshaft A.

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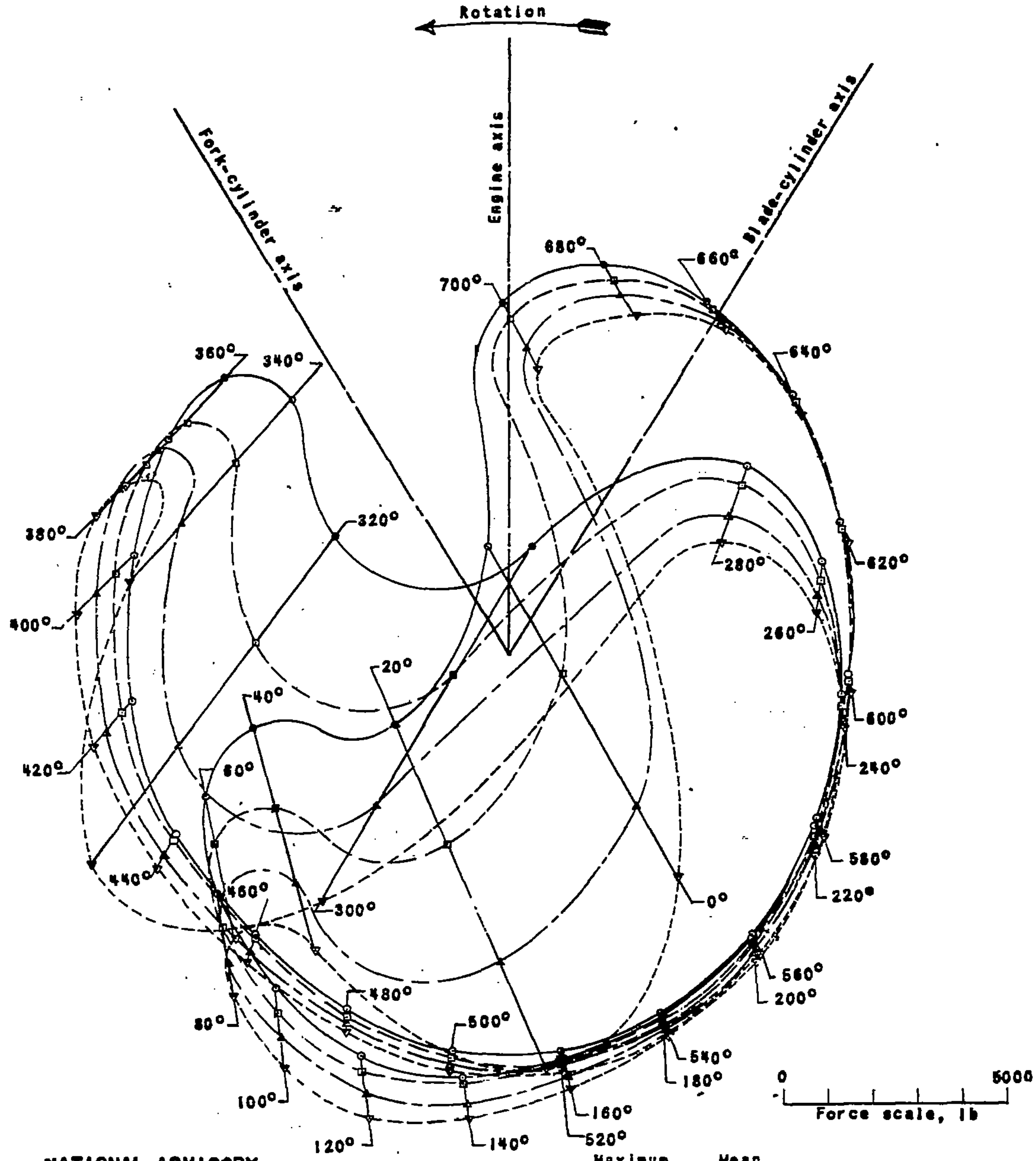

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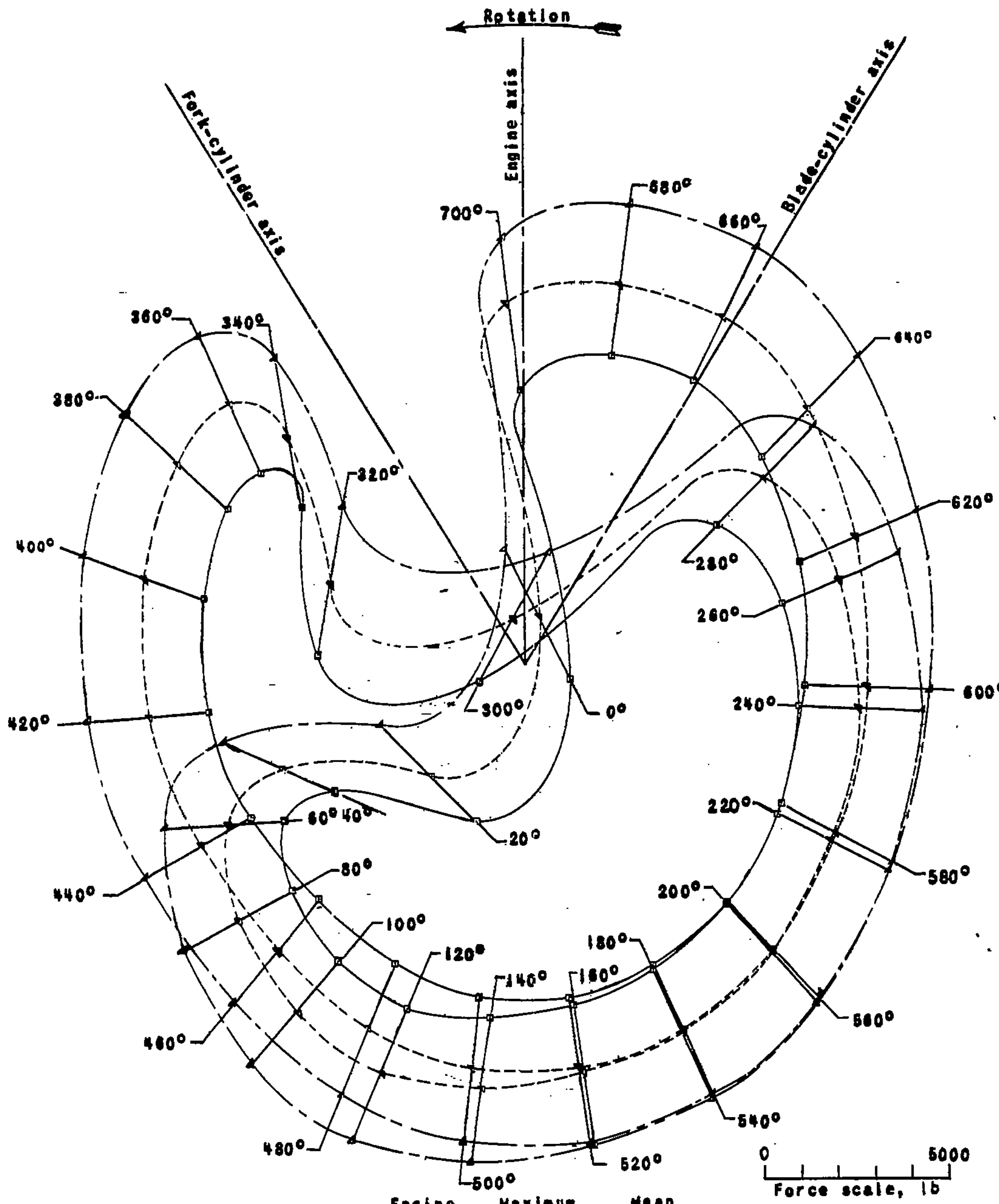
	Imep (lb/sq in.)	Maximum force (lb)	Mean force (lb)
—o—o—o—	182	10,240	8,260
—□—□—□—	242	10,740	8,470
—△—△—△—	303	11,320	8,630
—▽—▽—▽—	363	11,560	8,760

Indicated crank-angle degrees are for main bearing 1; add 360° in direction of rotation for main bearing 7.

(a) Engine speed, 3000 rpm; indicated mean effective pressures, 182, 242, 303, and 363 pounds per square inch.

Figure 9. - Polar diagrams showing the magnitude of the resultant force on end main bearings 1 and 7 of a V-type engine with crankshaft A and its direction with respect to the engine axis at different power conditions.



Engine speed (rpm)	Maximum force (lb)	Mean force (lb)
3000	10,740	8,470
3300	12,680	9,990
3600	14,660	12,840

Indicated crank-angle degrees are for main bearing 1; add 360° in direction of rotation for main bearing 7.

(b) Engine speeds, 3000, 3300, and 3600 rpm; indicated mean effective pressure, 242 pounds per square inch.

Figure 9. - Concluded.

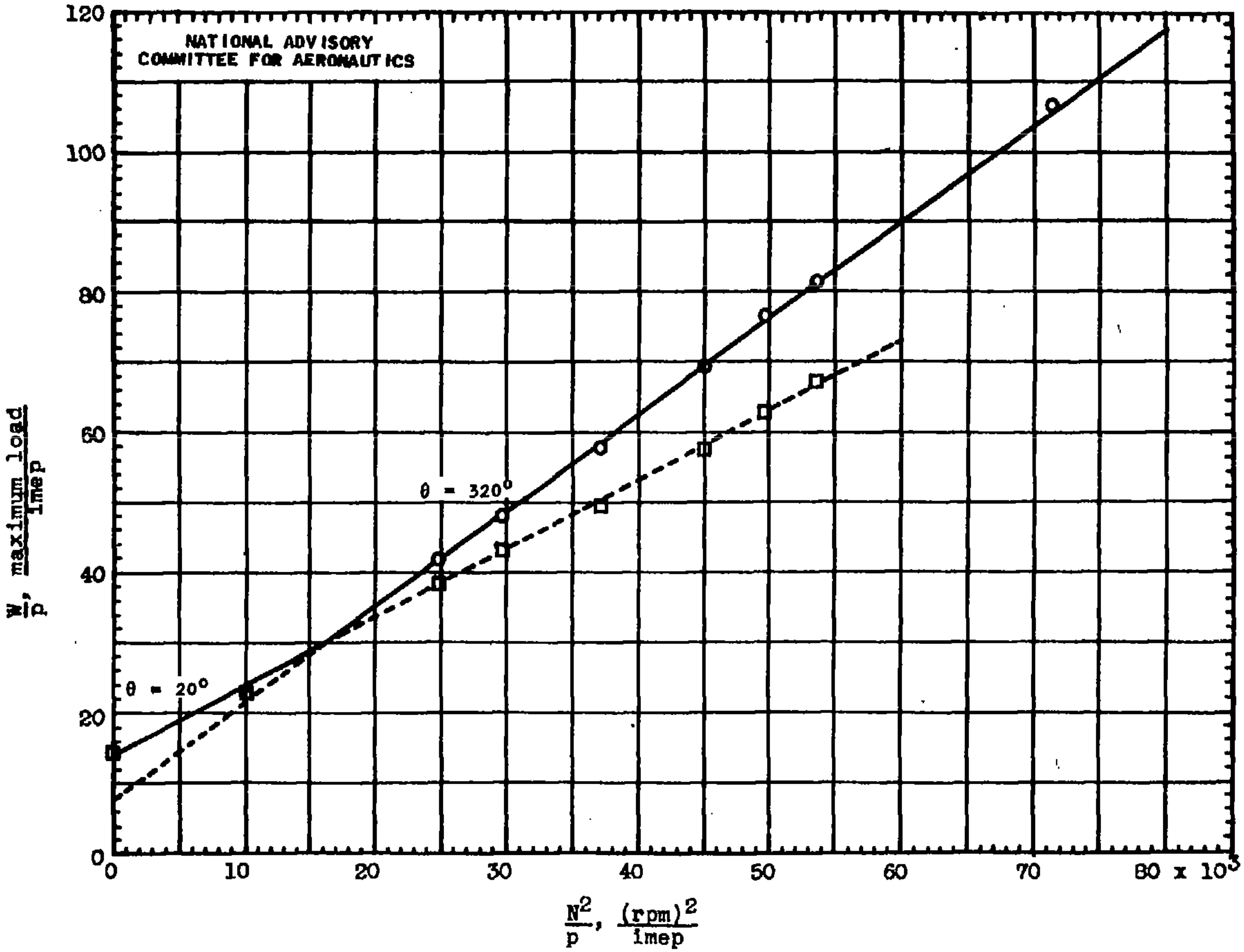


Figure 10. - Variation of W/p with N^2/p for the center main bearing of a production, V-type engine with crankshaft A at a compression ratio of 6.65.

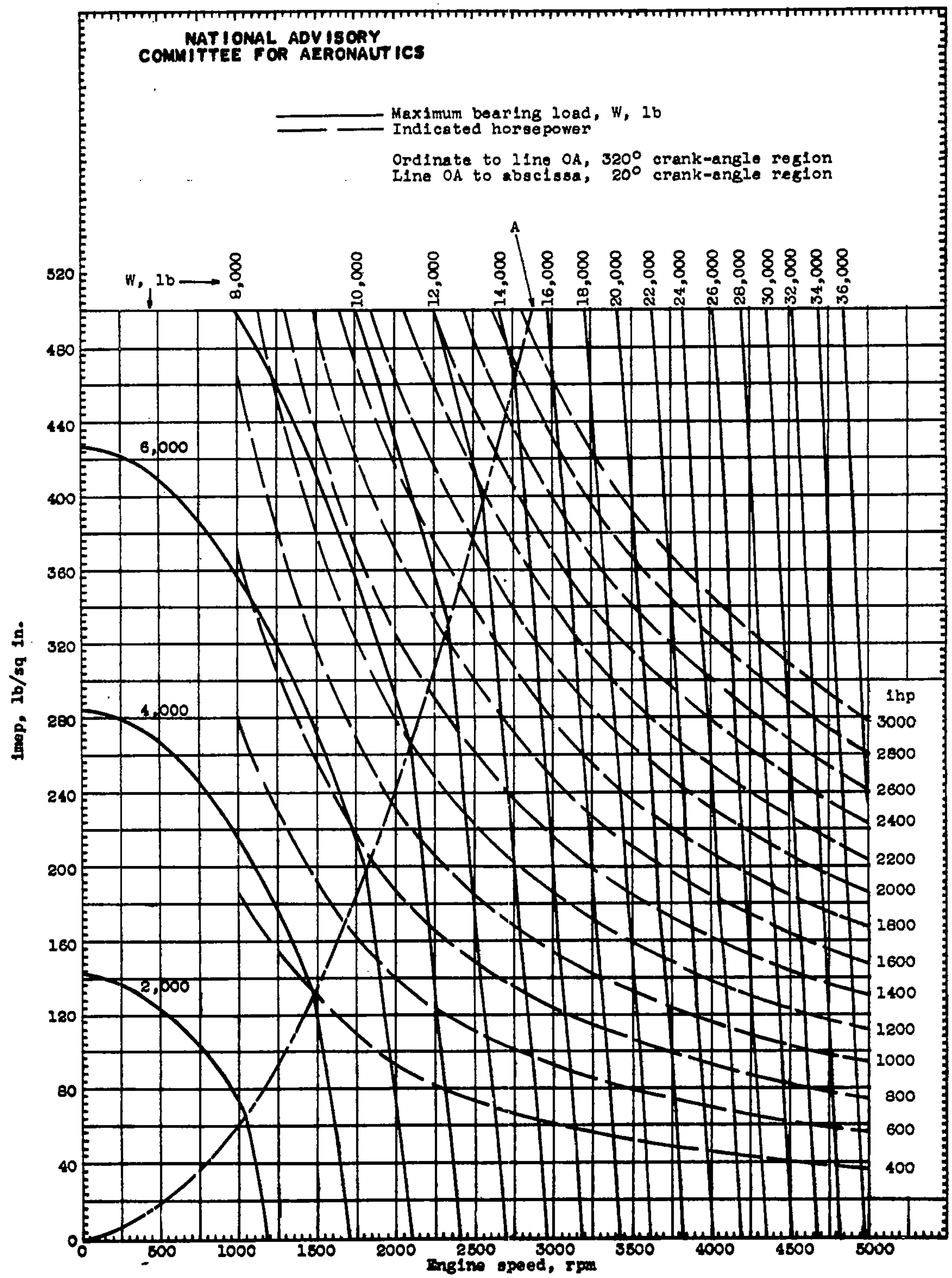


Figure 11. - Maximum load on center main bearing of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.

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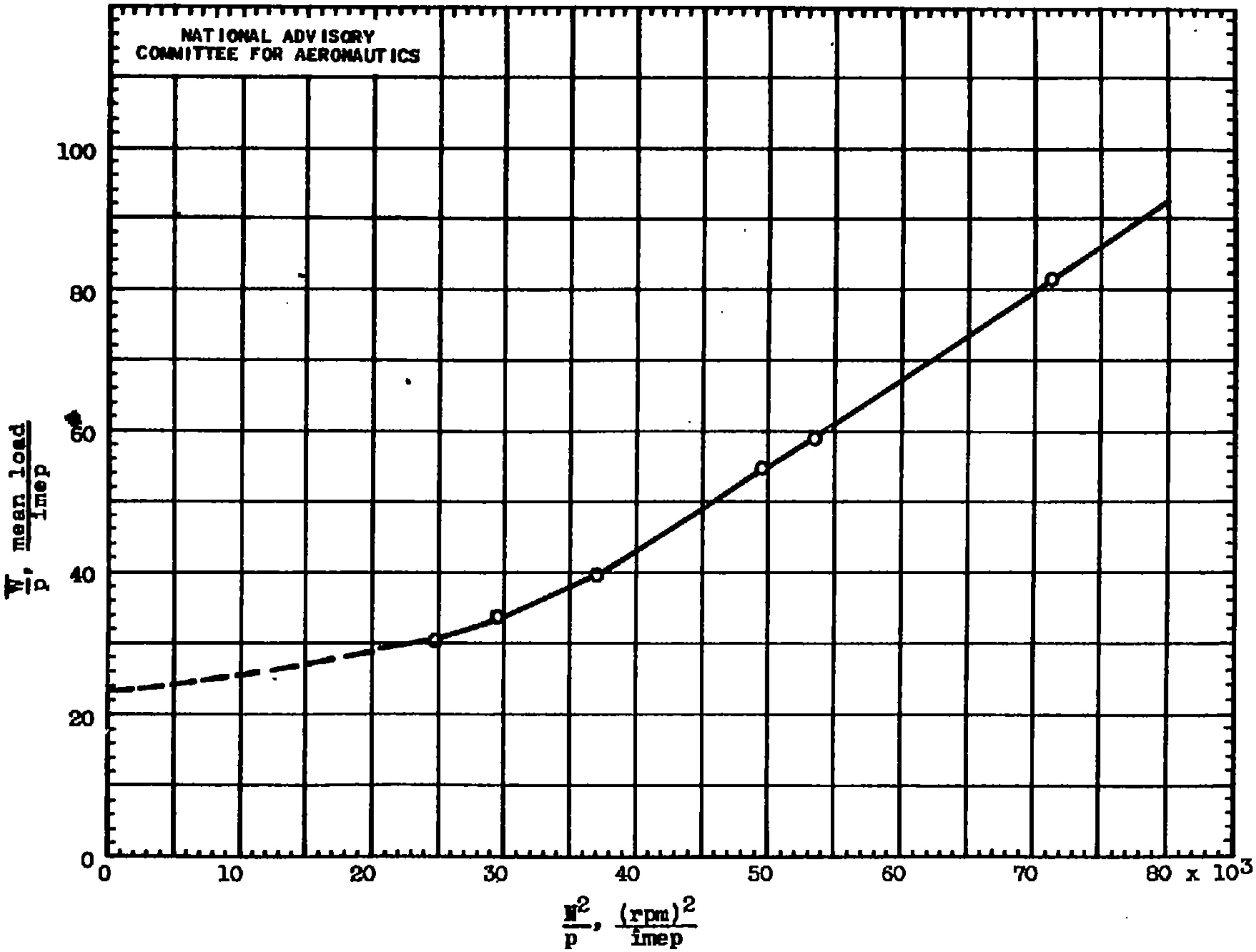

Figure 12. - Variation of $\overline{W}/p$ with N^2/p for the center main bearing of a production, V-type engine with crankshaft A at a compression ratio of 6.65.

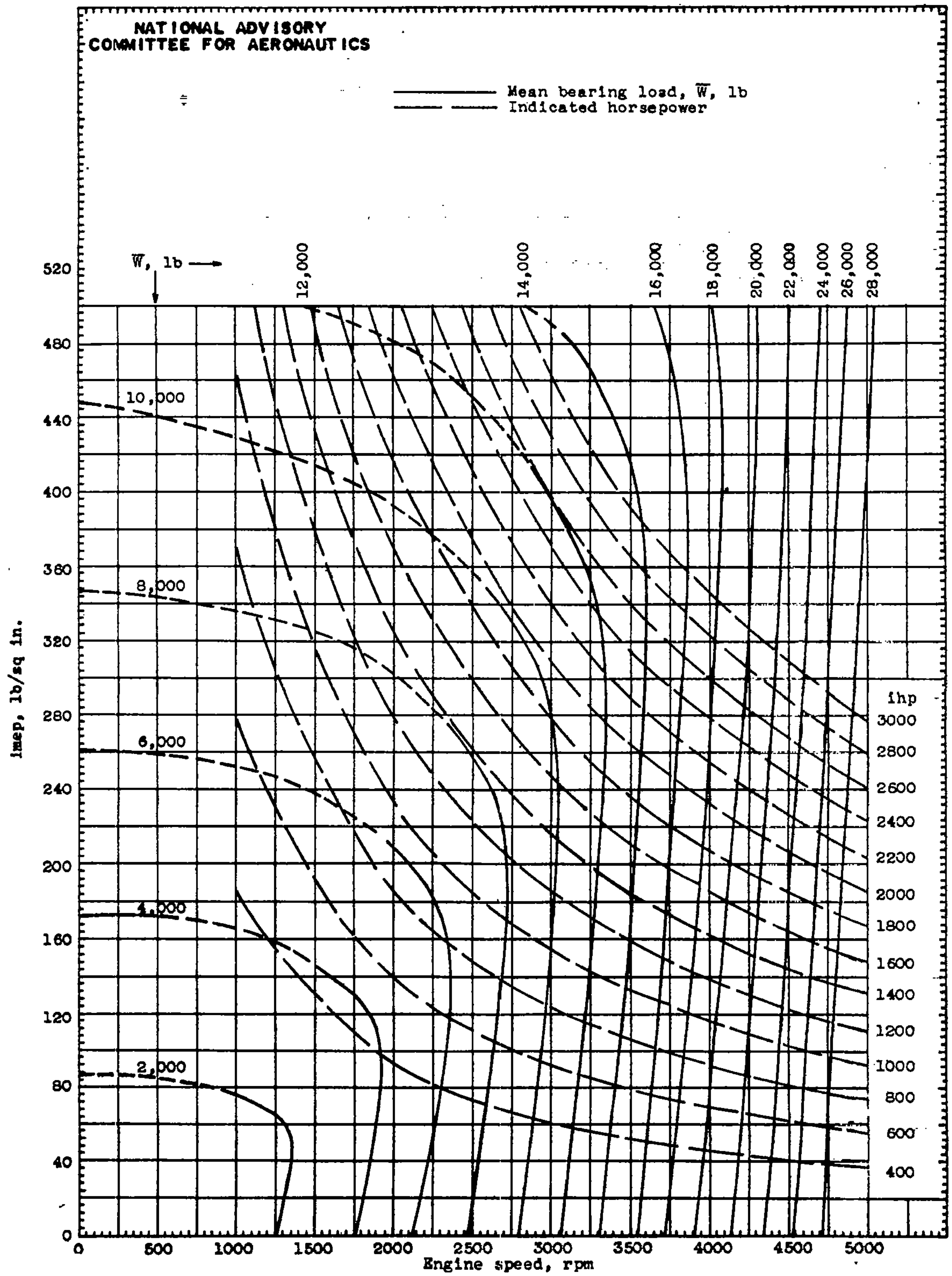

Figure 13. - Mean load on center main bearing of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed. (Constant mean-load curves.) Effective bearing area, 7.12 square inches.

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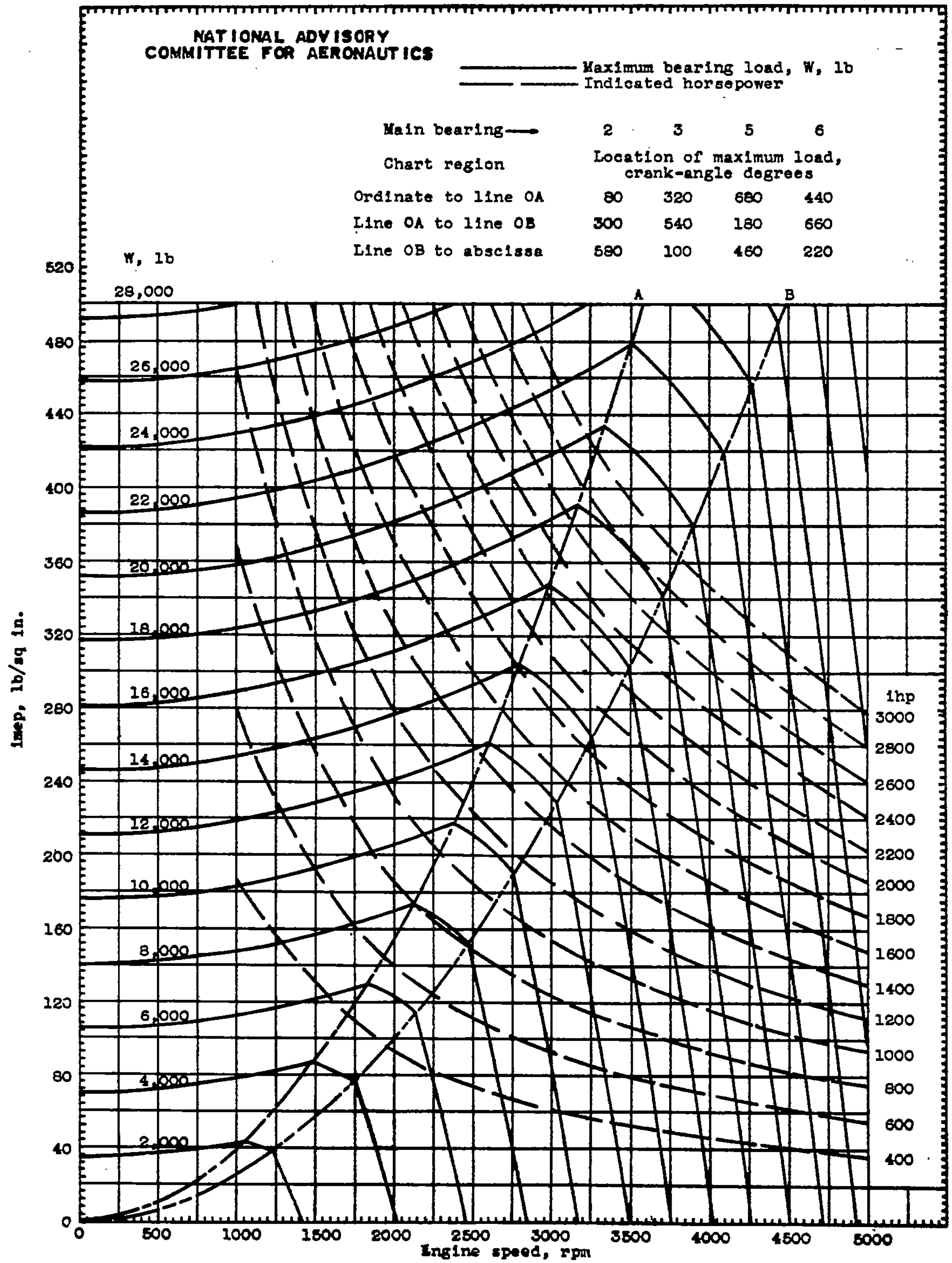

Figure 14. - Maximum load on intermediate main bearings 2, 3, 5, and 6 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.

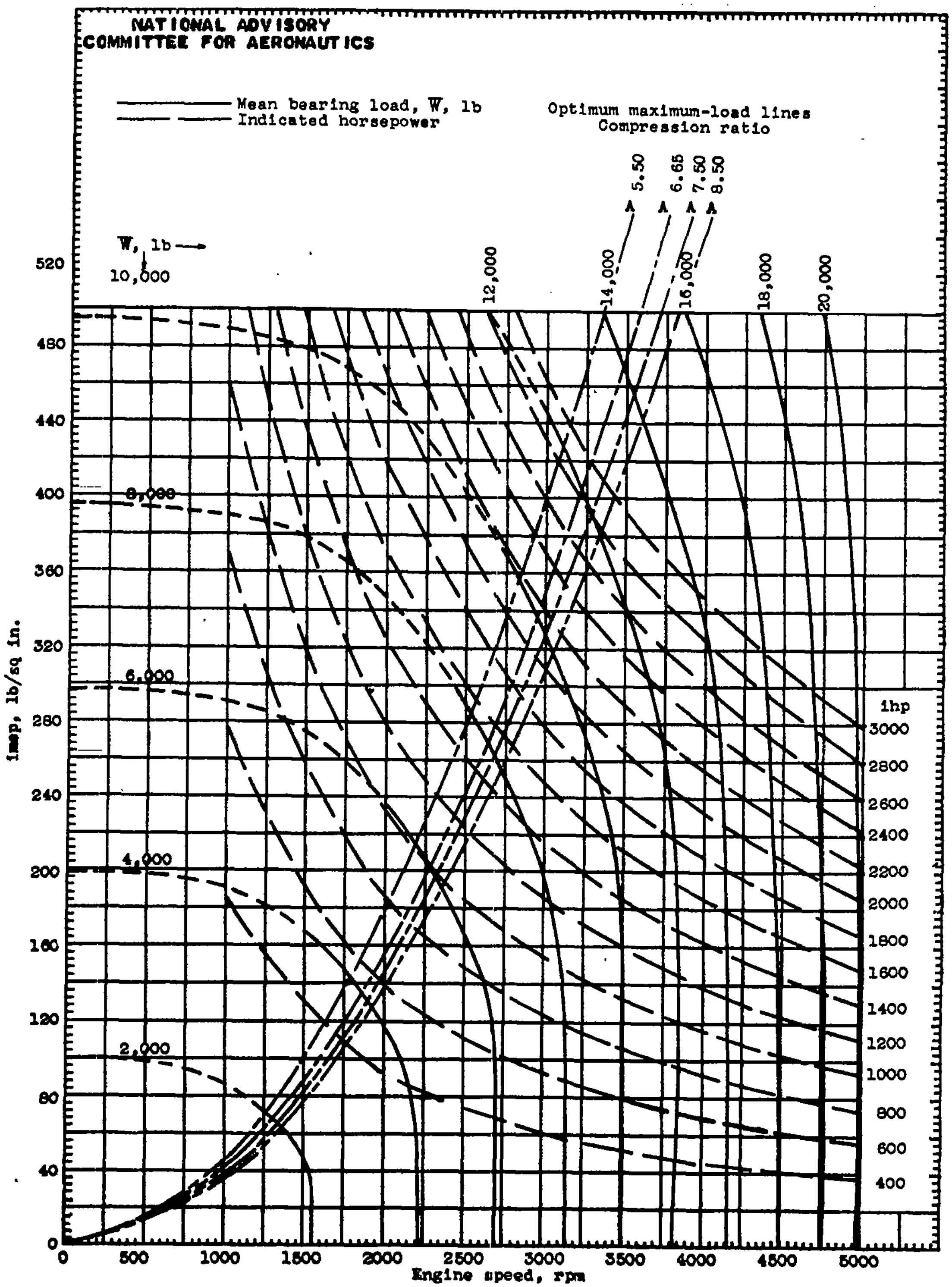

Figure 15. - Mean load on intermediate main bearings 2, 3, 5, and 6 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed. (Constant mean-load curves.) Effective bearing area, 4.13 square inches.

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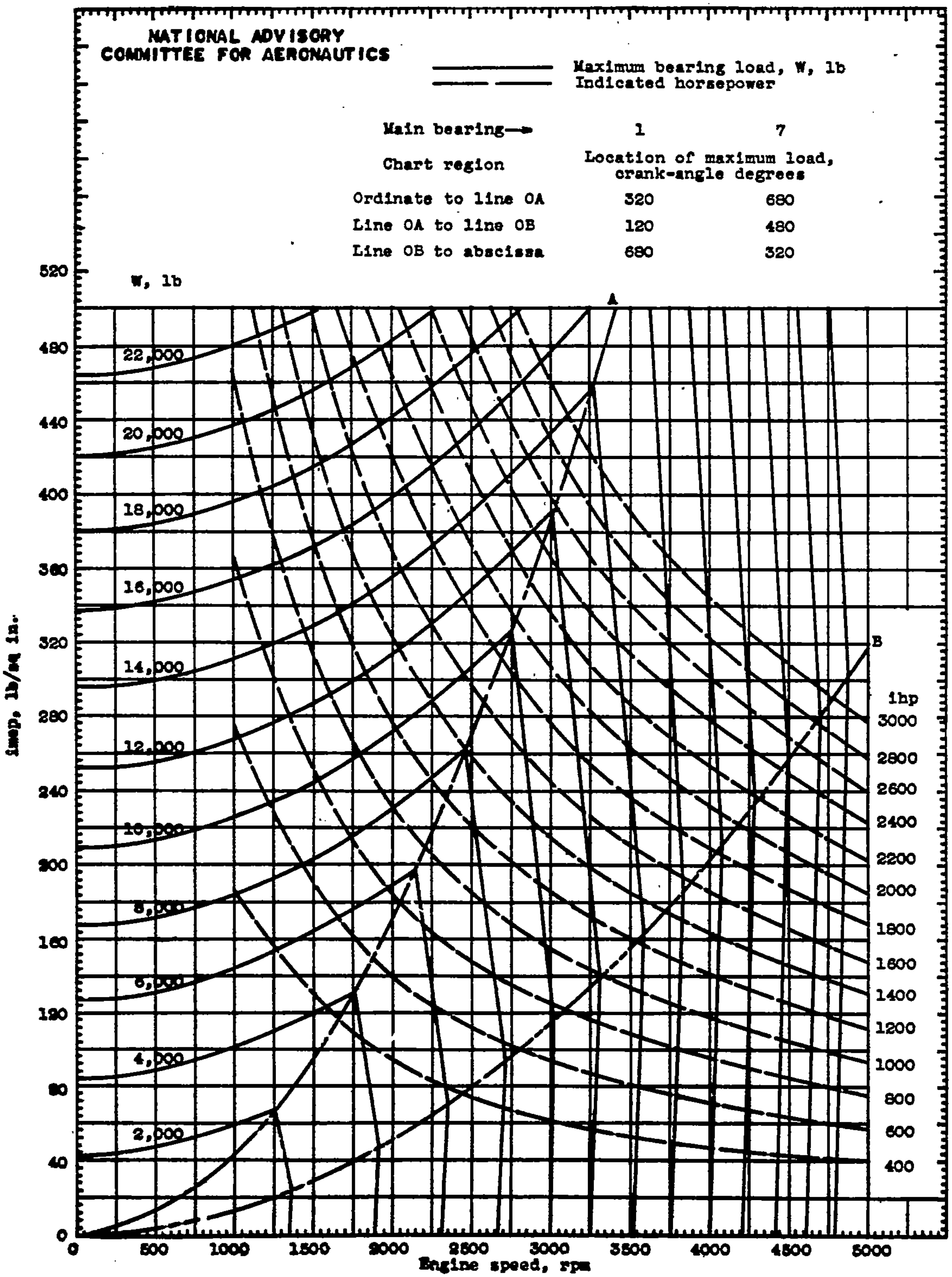

Figure 16. - Maximum load on end main bearings 1 and 7 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.

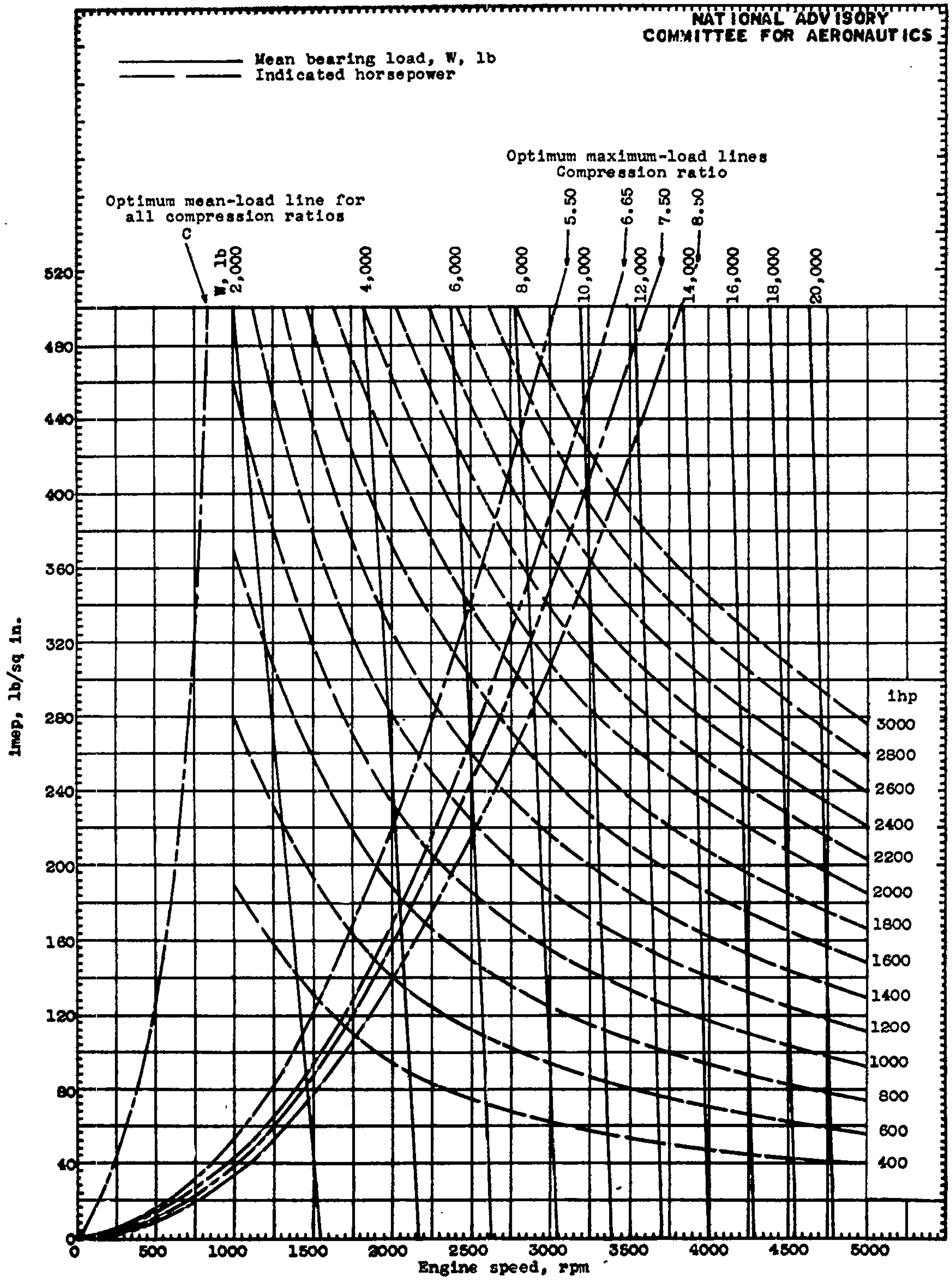

Figure 17. - Mean load on end main bearings 1 and 7 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed. (Constant mean-load curves.) Effective bearing area, 4.13 square inches.

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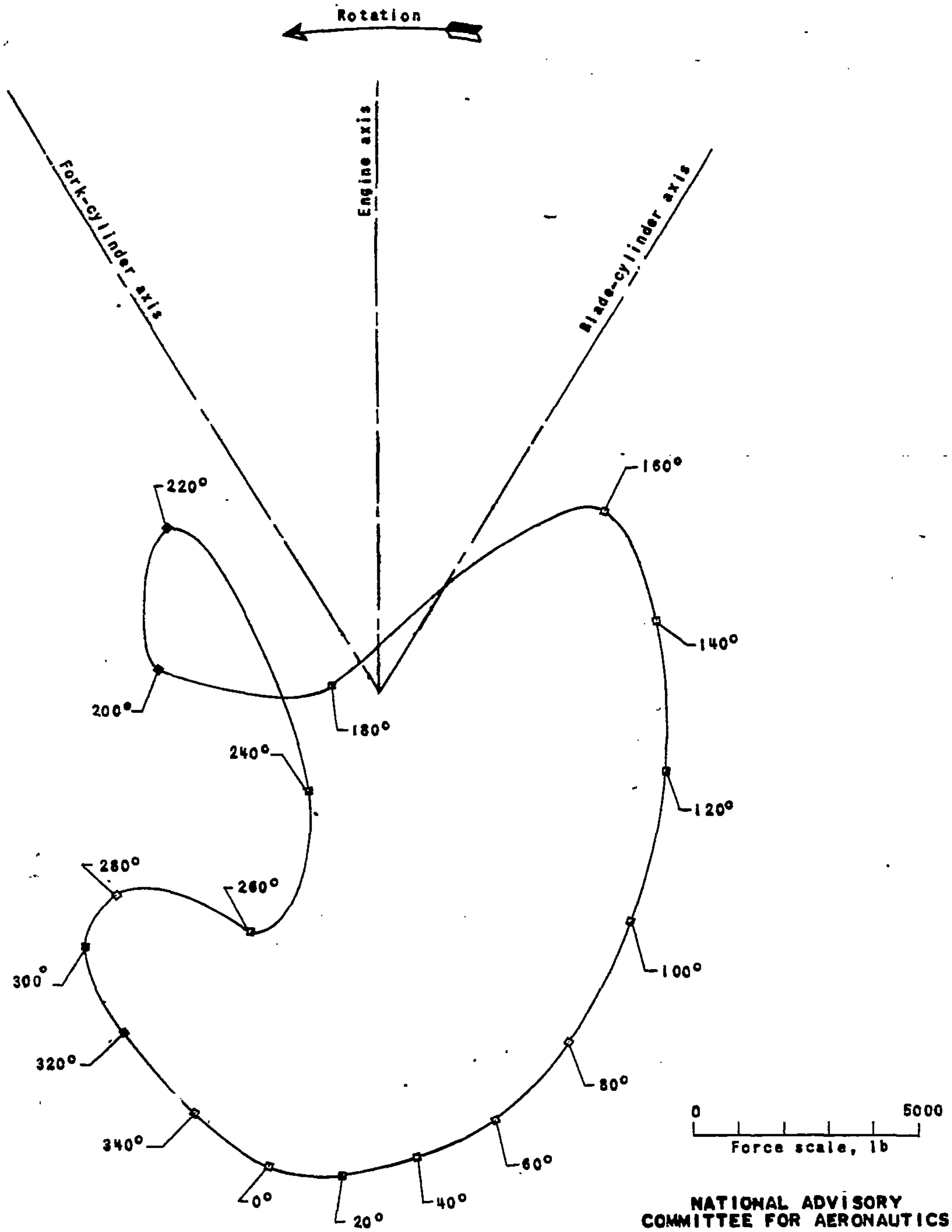

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Figure 18. - Polar diagram showing the magnitude of the resultant force on the center main bearing of a production, V-type engine with crankshaft B and its direction with respect to the engine axis. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.

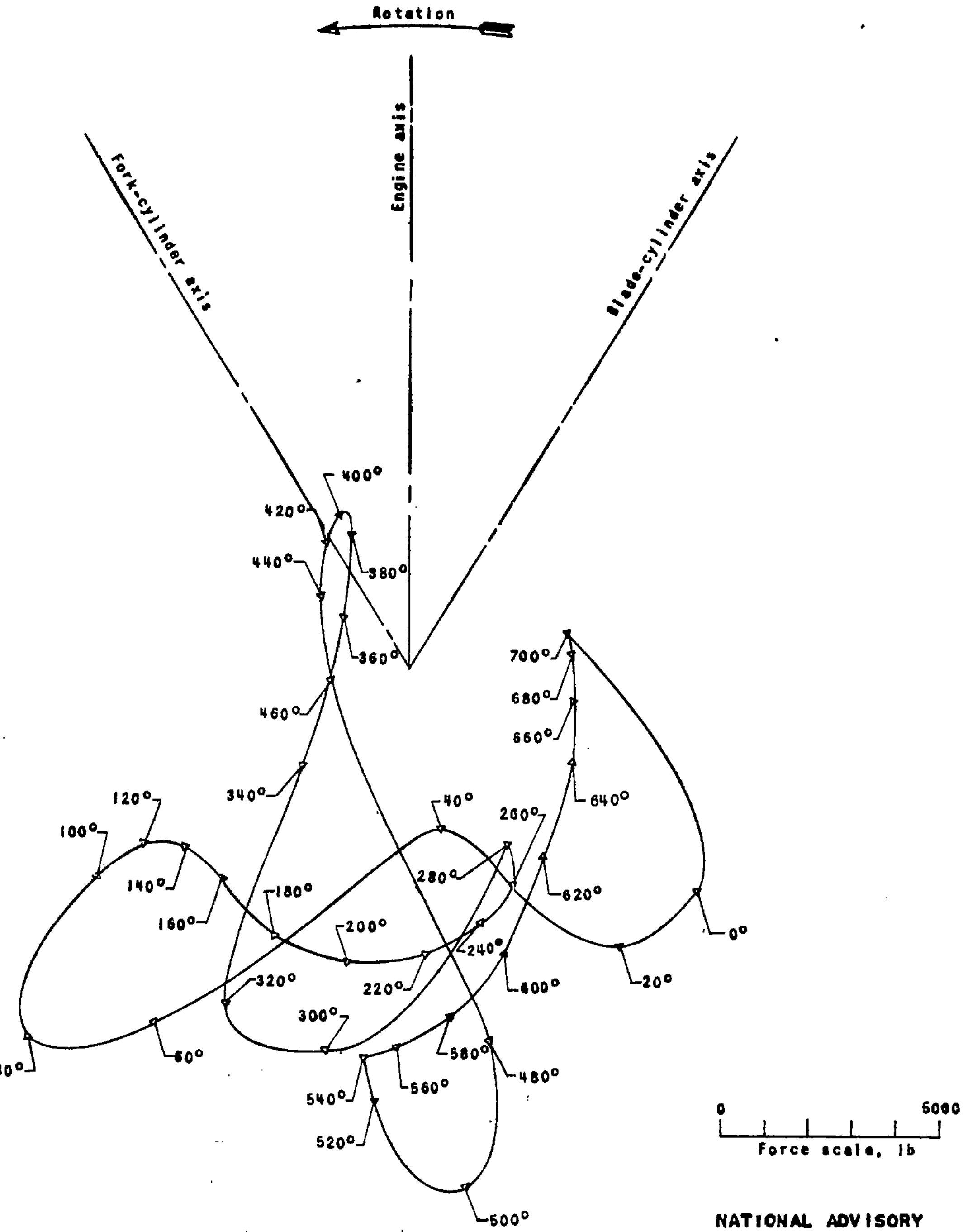

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Figure 19. - Polar diagram showing the magnitude of the resultant force on intermediate main bearing 2 of a production, V-type engine with crankshaft B and its direction with respect to the engine axis. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.

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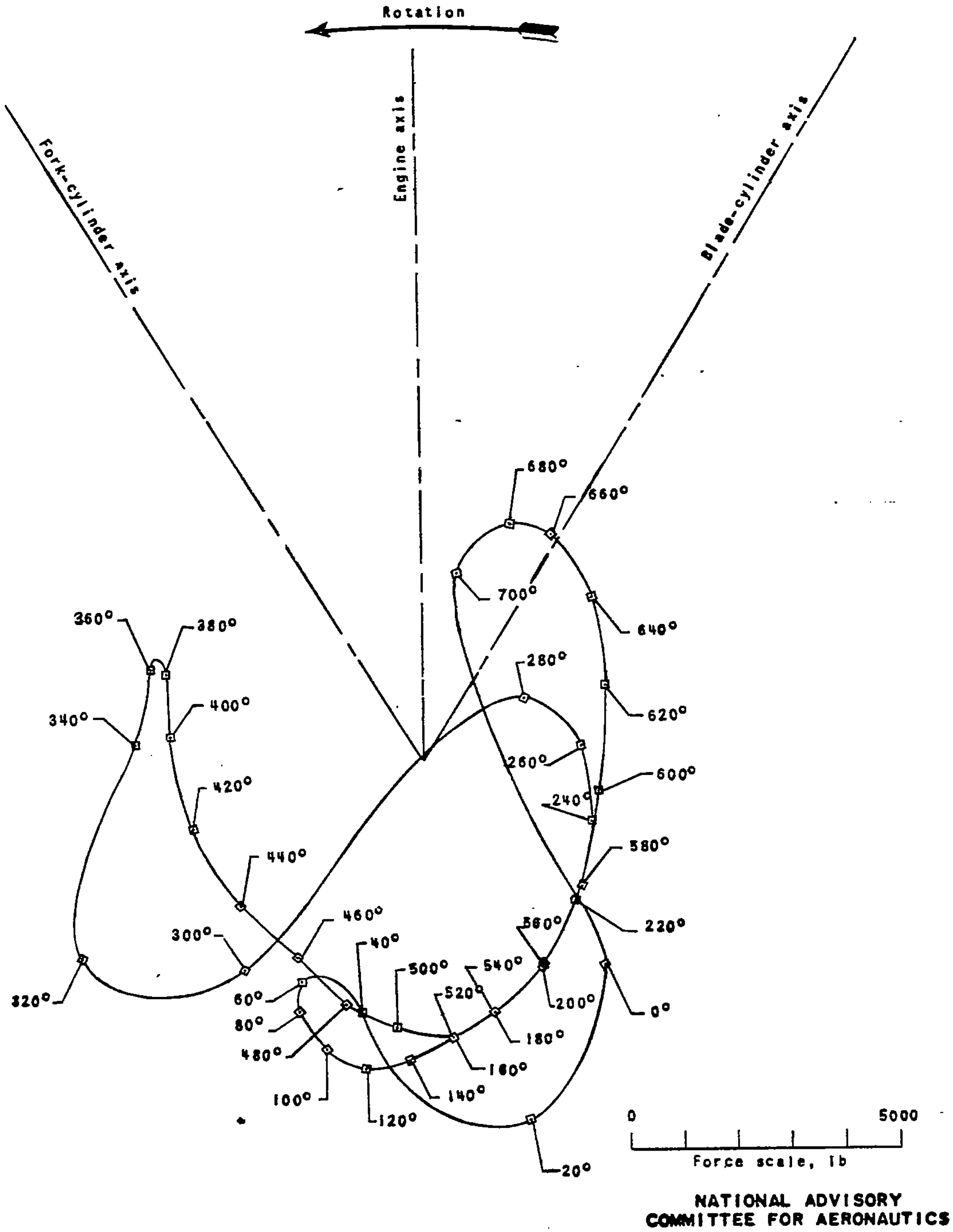

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Figure 20. - Polar diagram showing the magnitude of the resultant force on end main bearing I of a production, V-type engine with crankshaft B and its direction with respect to the engine axis. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.

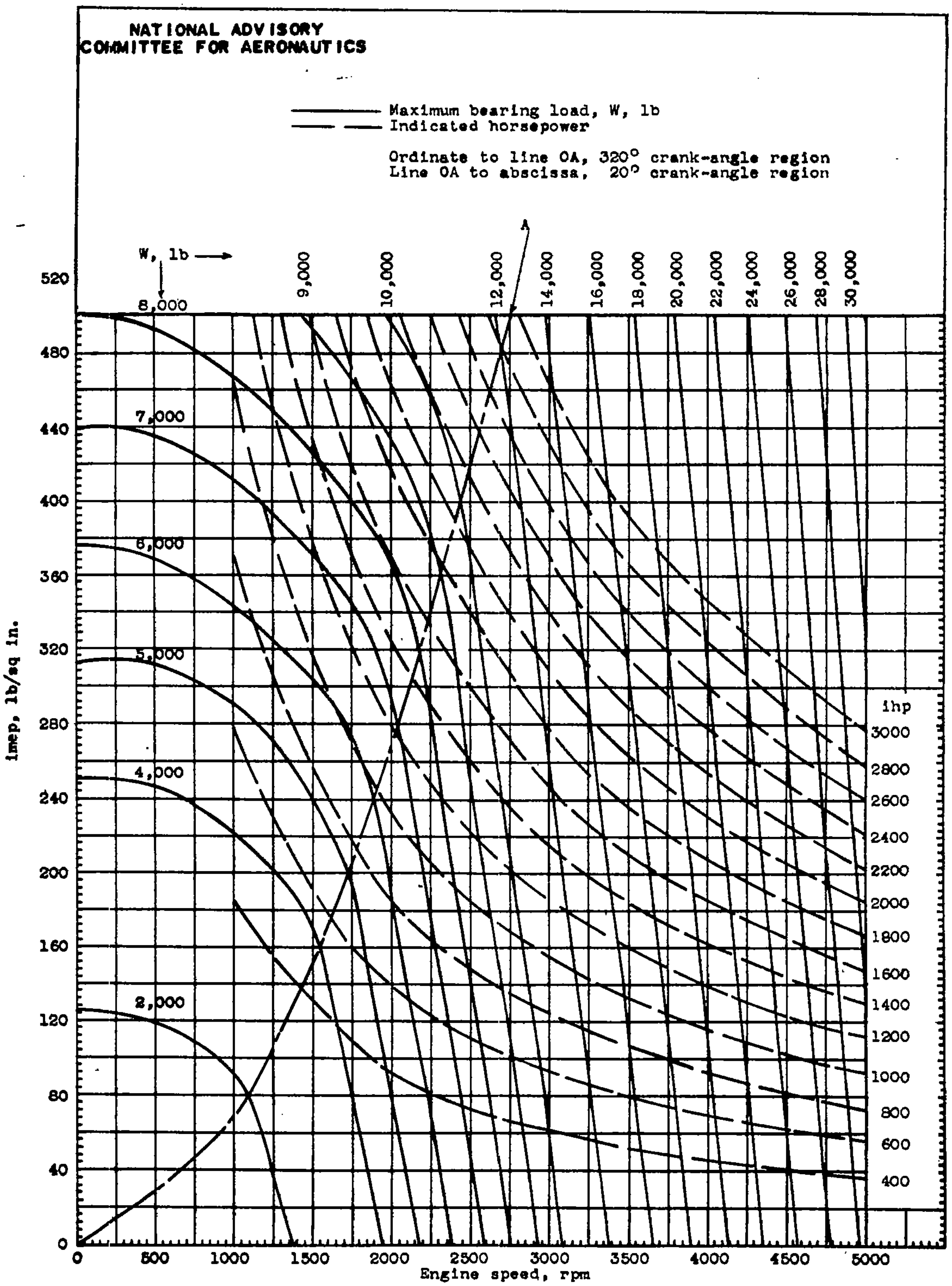

Figure 21. - Maximum load on center main bearing of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 7.12 square inches.

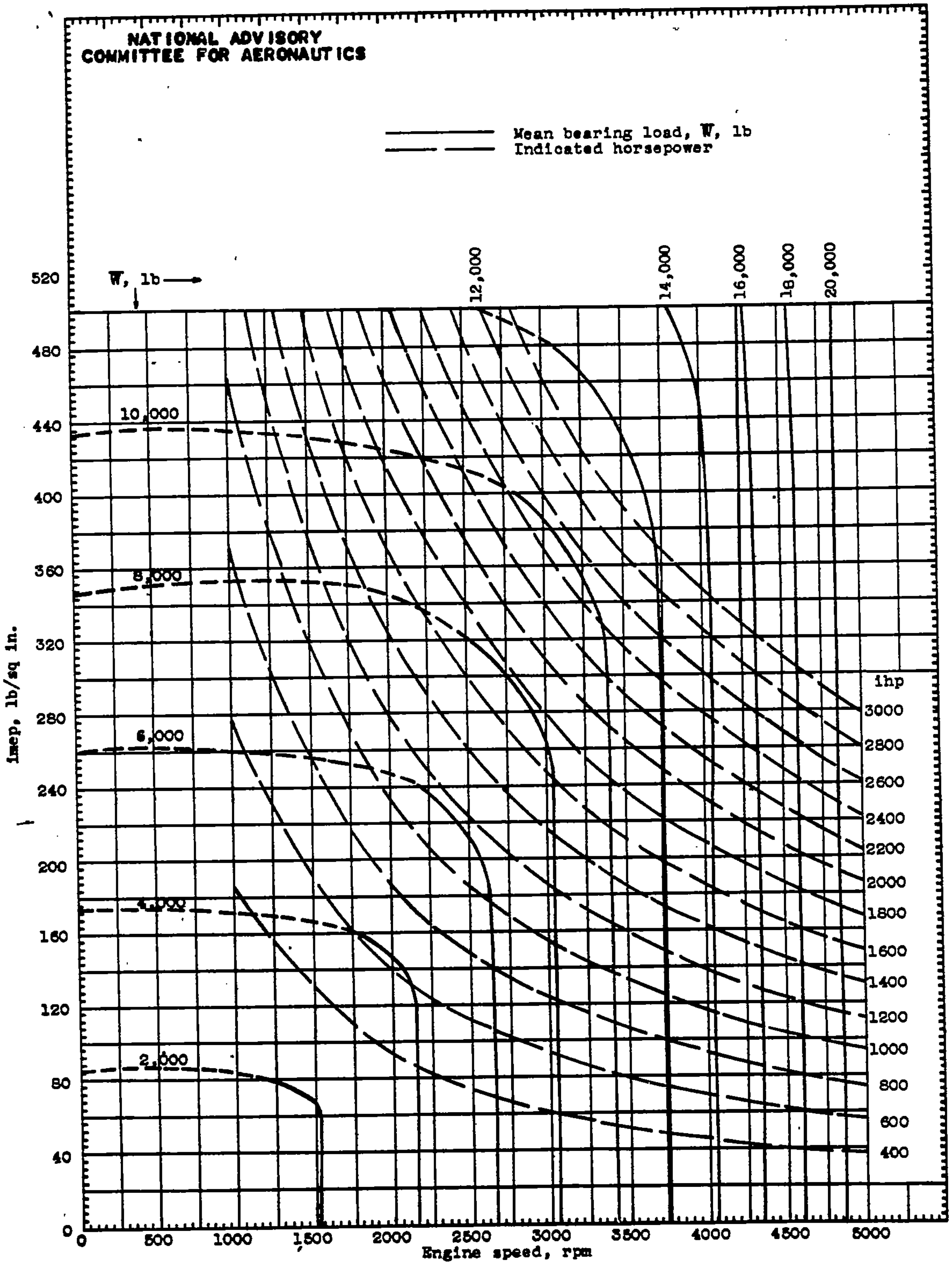

Figure 22. - Mean load on center main bearing of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant mean-load curves.) Effective bearing area, 7.12 square inches.

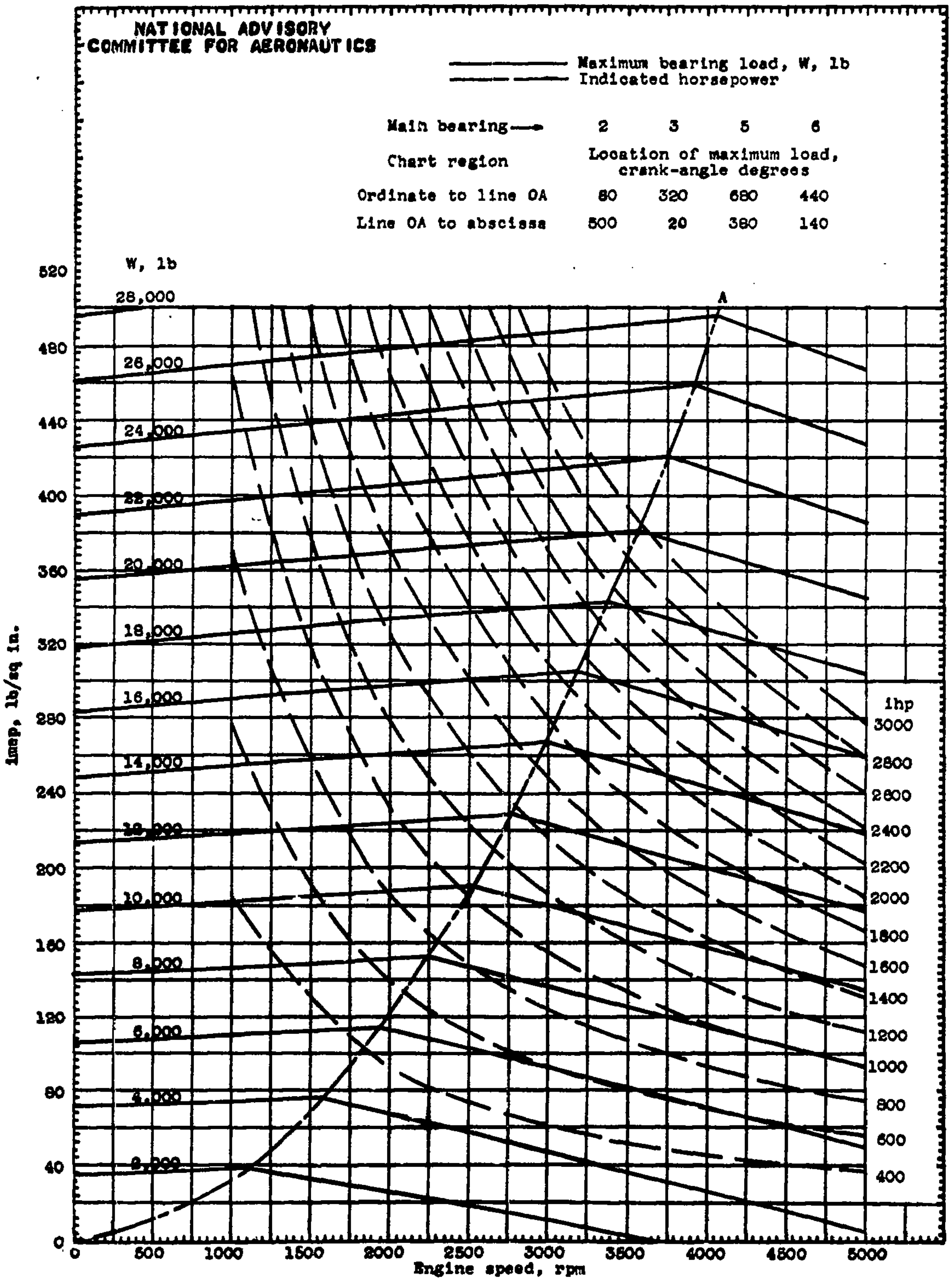

Figure 23. - Maximum load on intermediate main bearings 2, 3, 5, and 6 of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.

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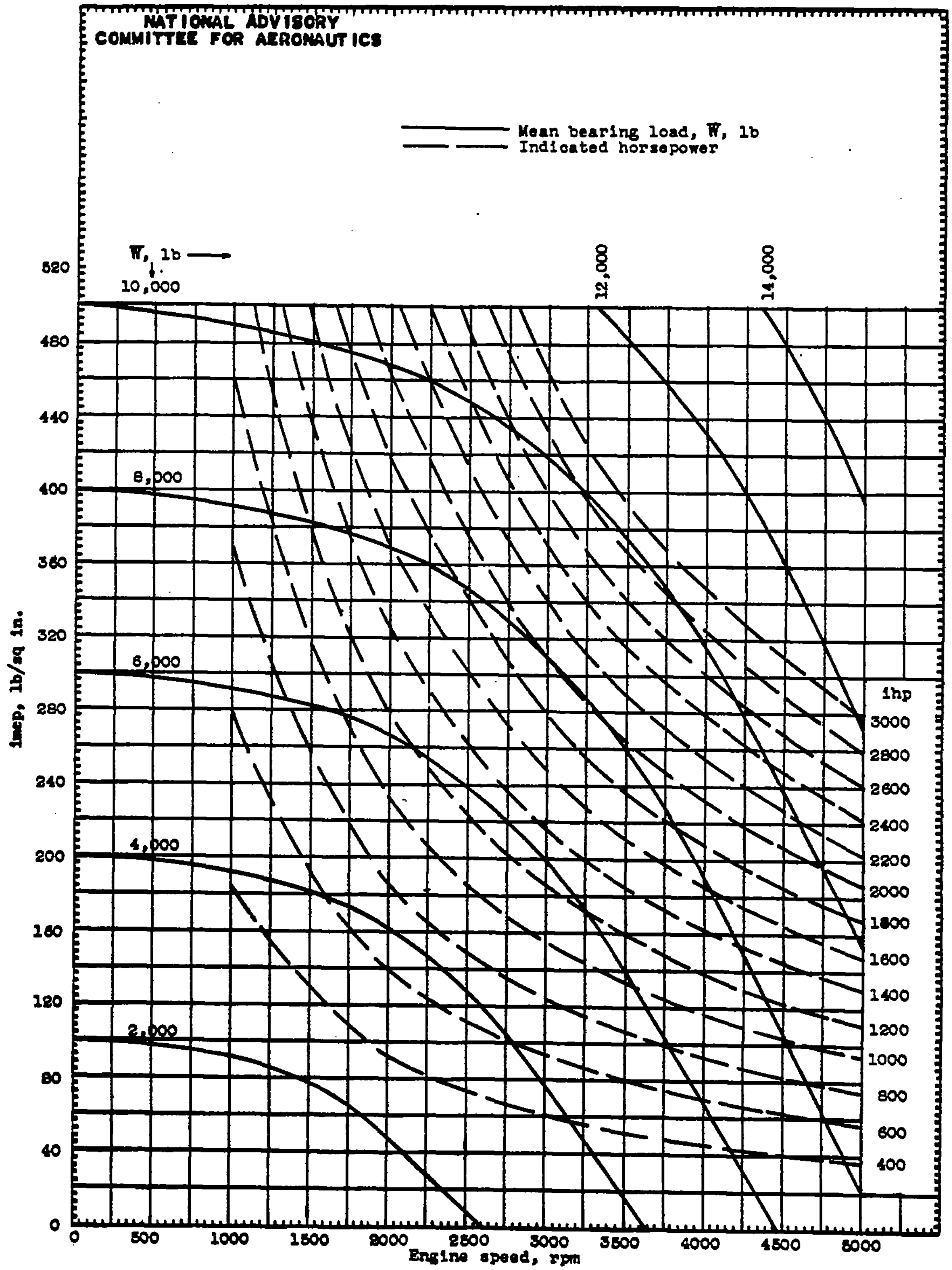

Figure 24. -Mean loads on intermediate main bearings 2, 3, 5, and 6 of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant mean-load curves.) Effective bearing area, 4.13 square inches.

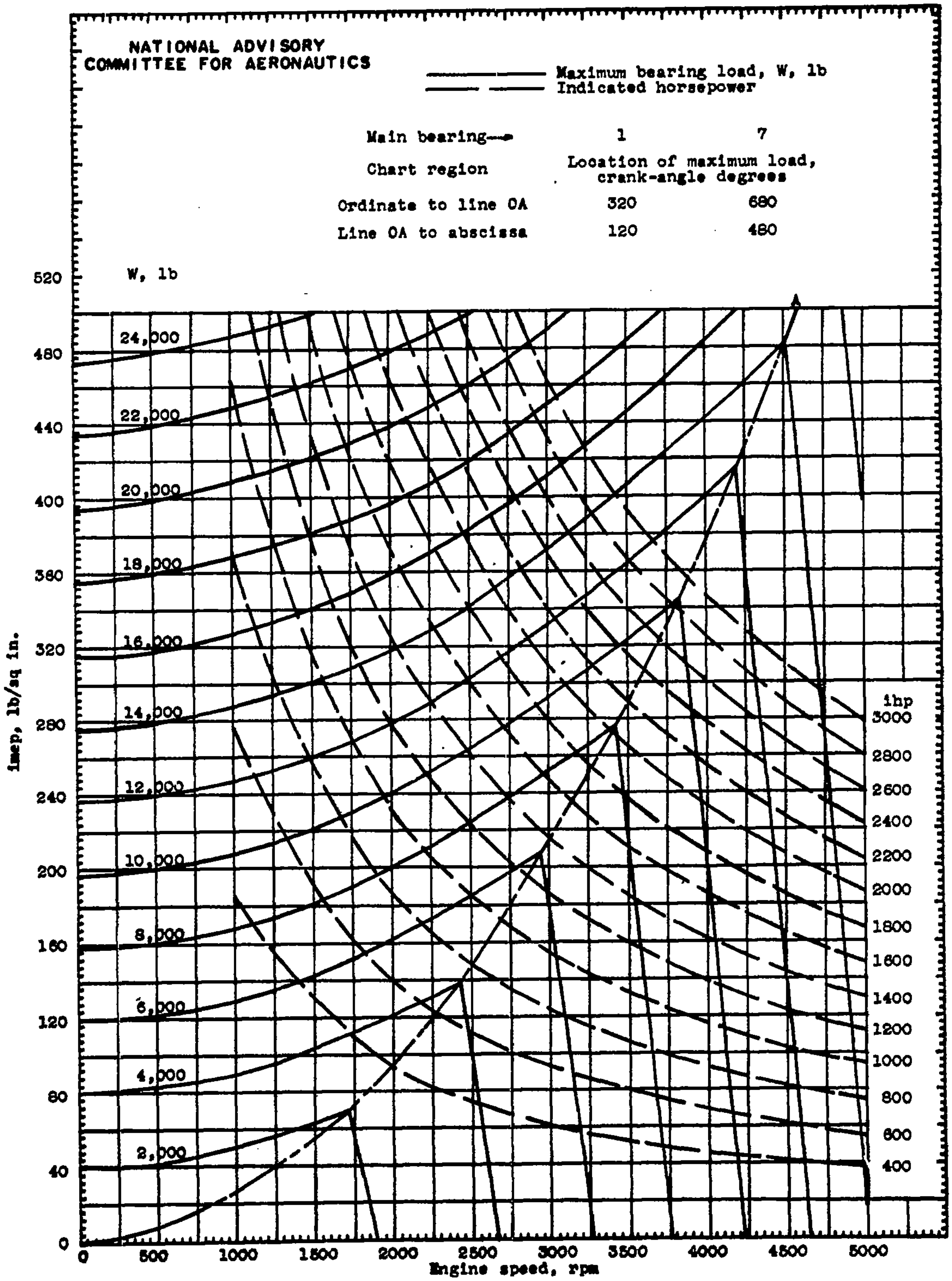

Figure 25. - Maximum load on end main bearings 1 and 7 of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.

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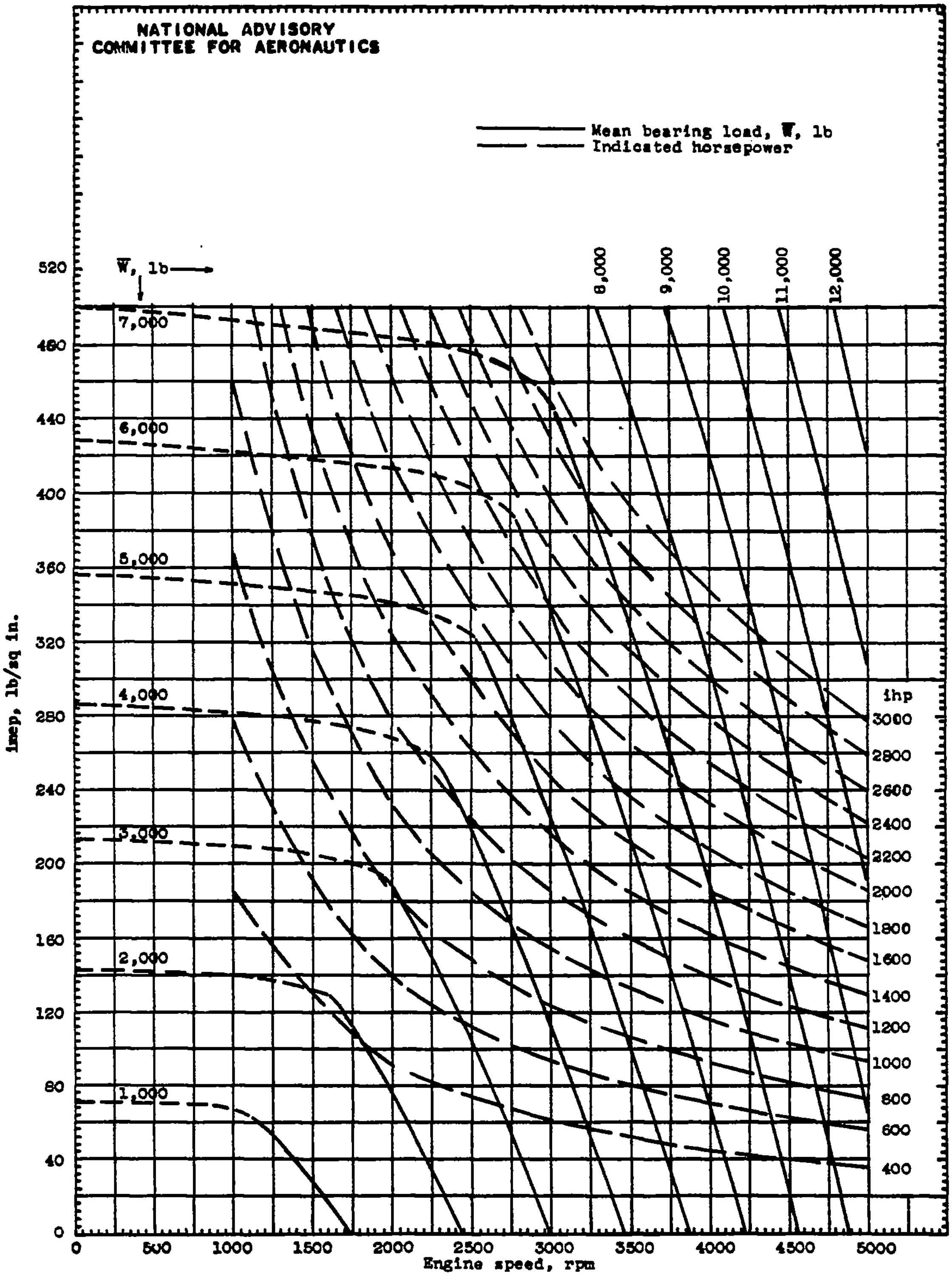
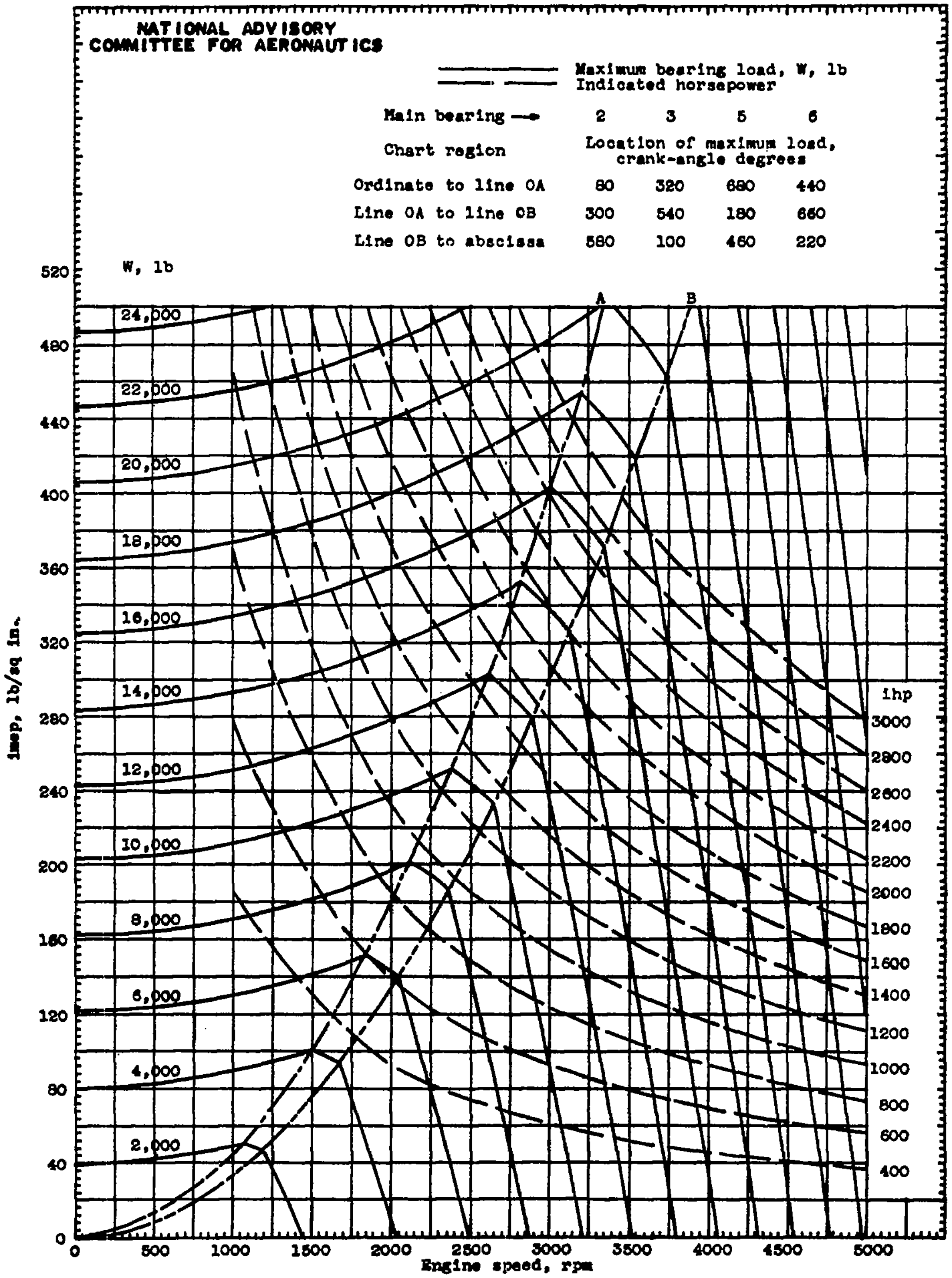
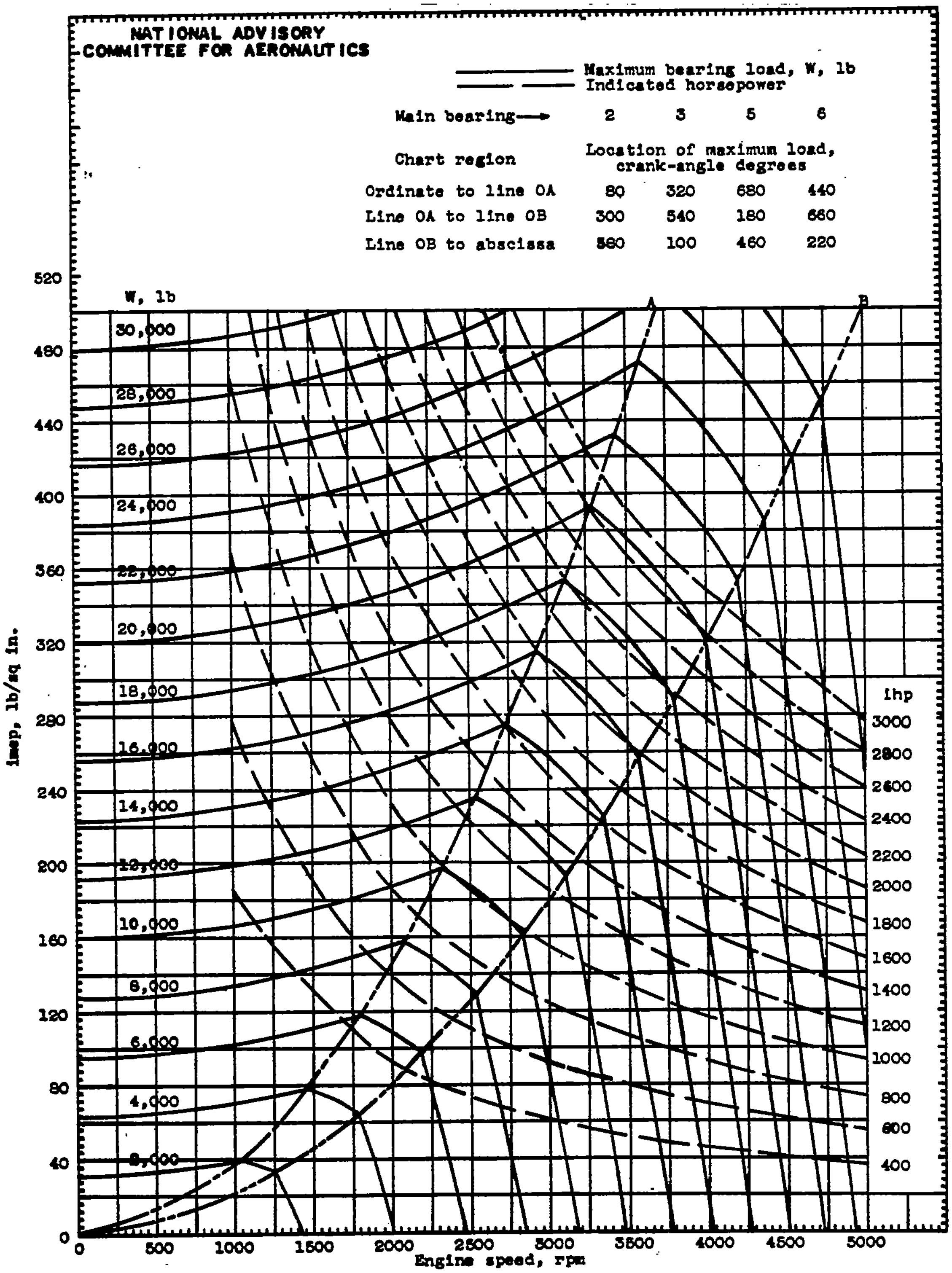


Figure 26. - Mean load on end main bearings 1 and 7 of a production, V-type engine with crankshaft B for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant mean-load curves.) Effective bearing area, 4.13 square inches.

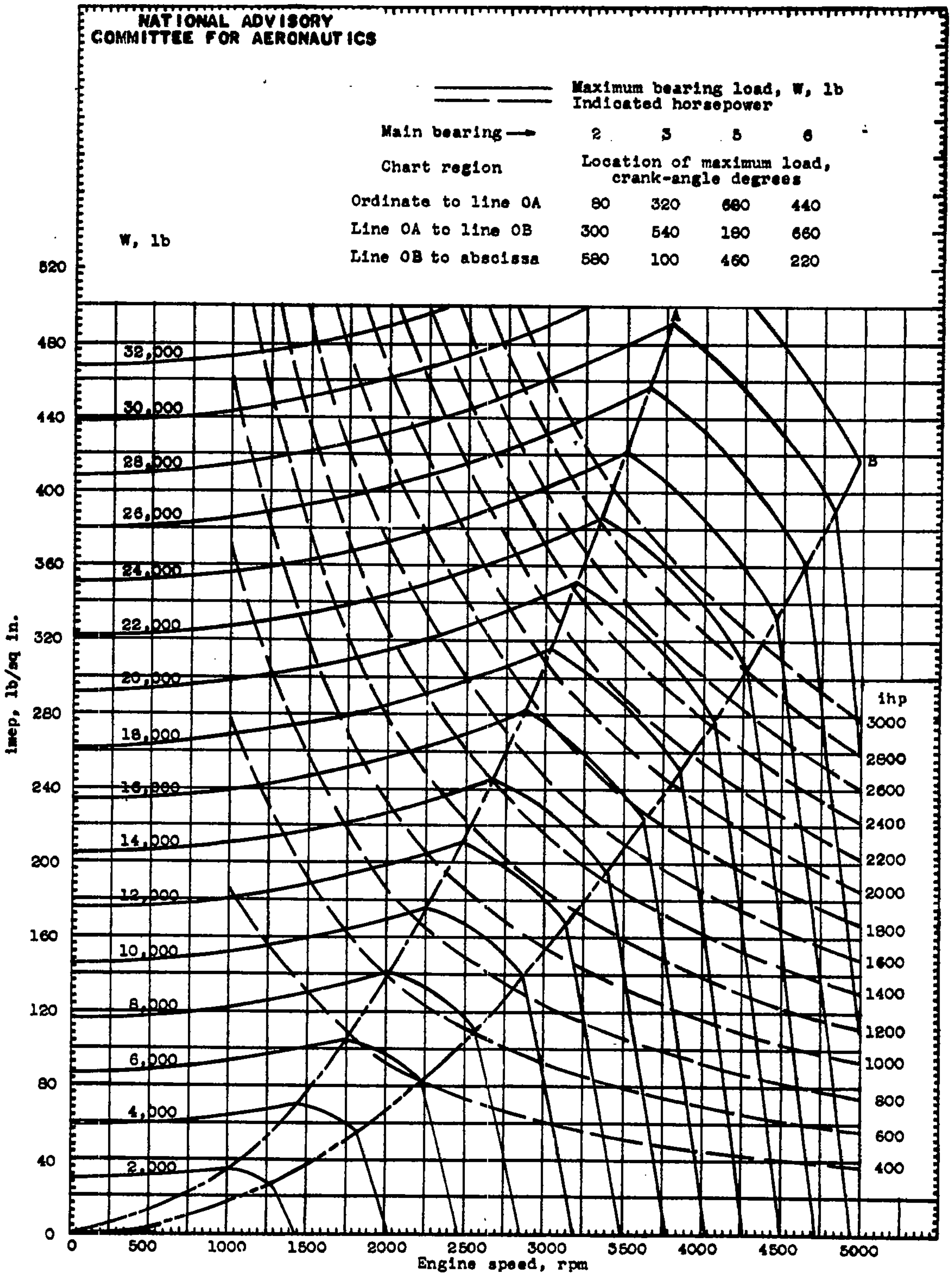


(a) Compression ratio, 5.50.

Figure 27. - Maximum load on intermediate main bearings 2, 3, 5, and 6 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed at various compression ratios. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.



(b) Compression ratio, 7.50.
Figure 27. - Continued.



(c) Compression ratio, 8.50.
Figure 27. - Concluded.

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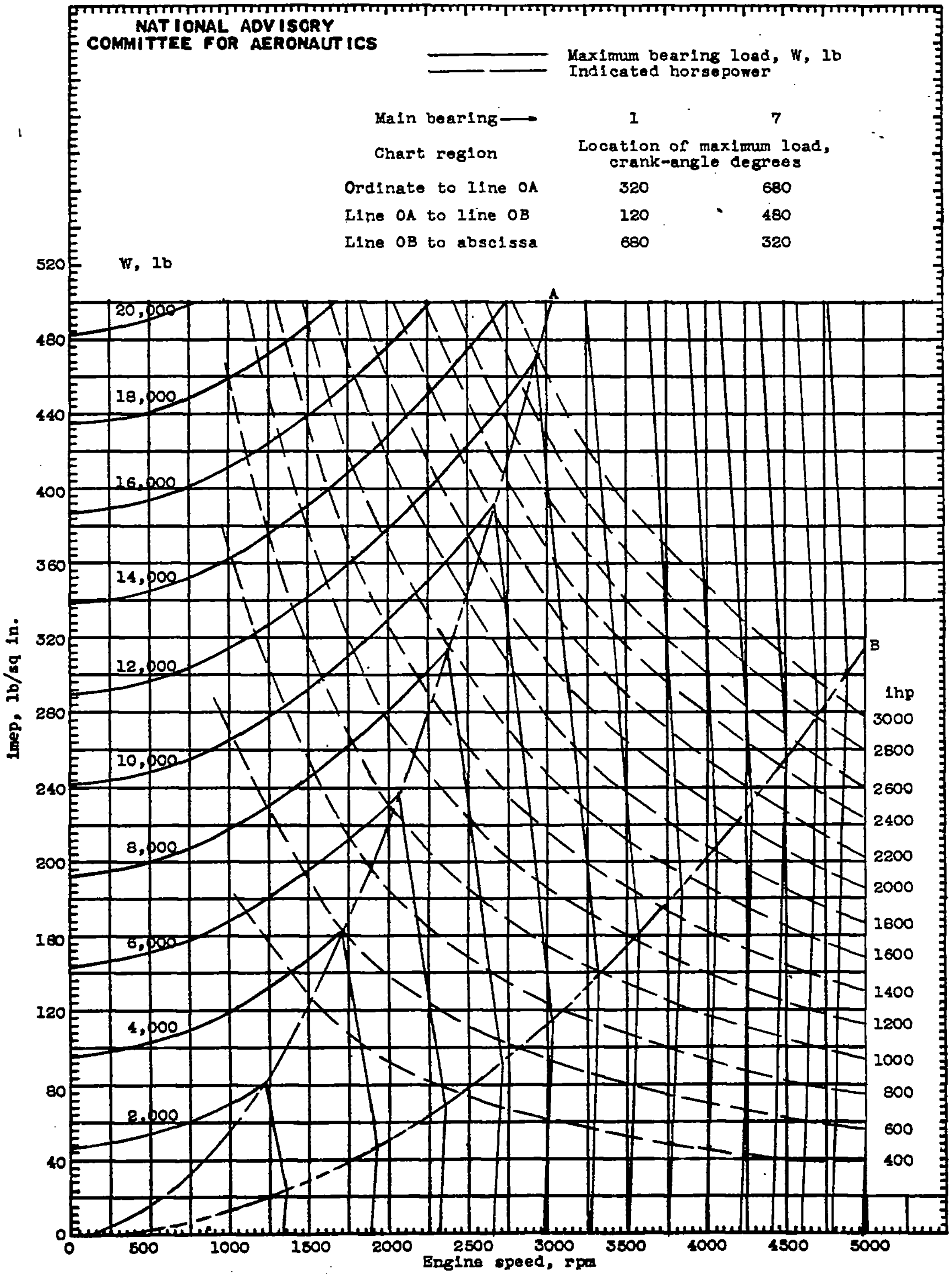
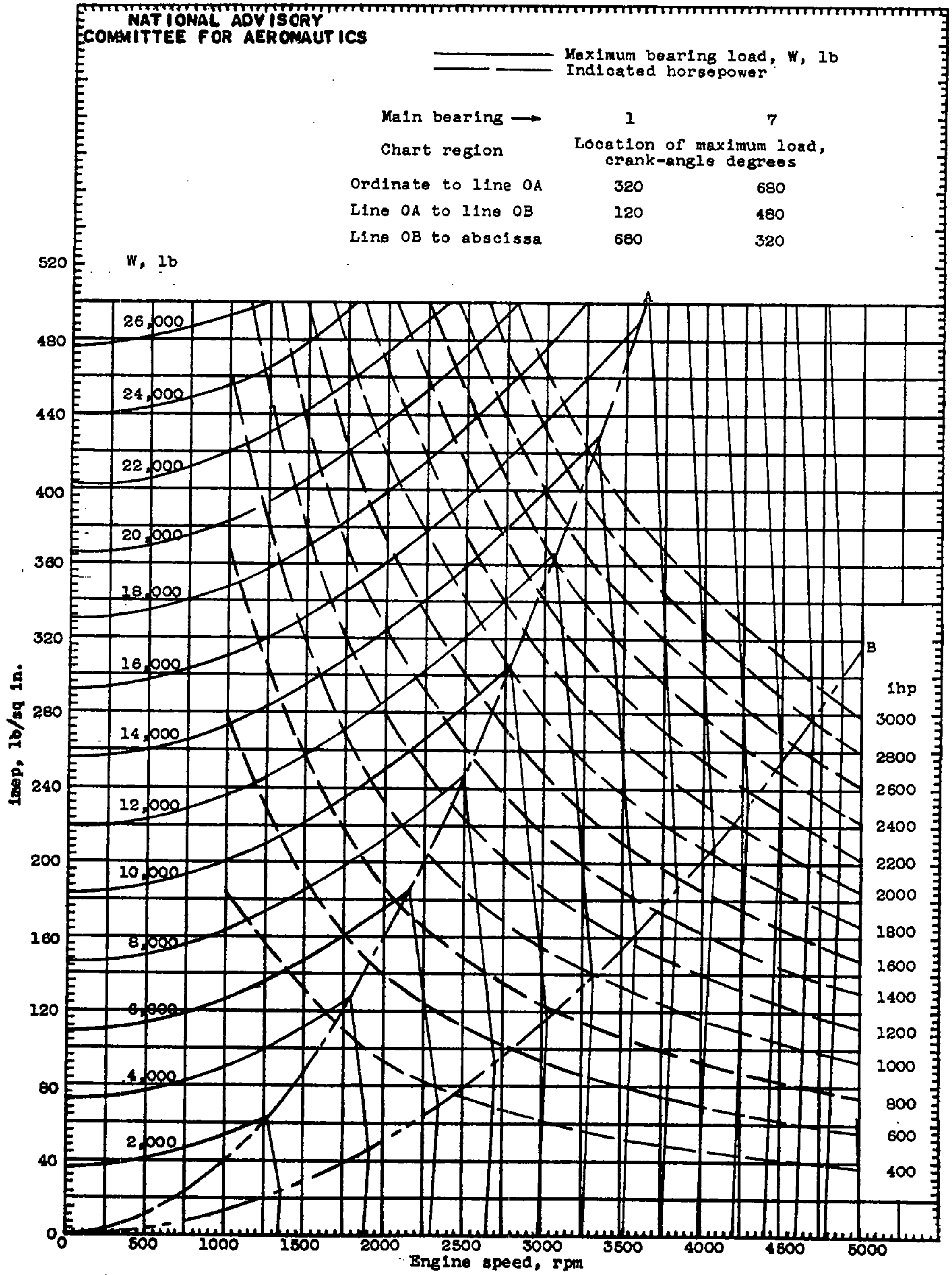


(a) Compression ratio, 5.50.

Figure 28. - Maximum load on end main bearings 1 and 7 of a production, V-type engine with crankshaft A for all values of indicated mean effective pressure and engine speed at various compression ratios. (Constant maximum-load curves.) Effective bearing area, 4.13 square inches.



(b) Compression ratio, 7.50.
Figure 28. - Continued.

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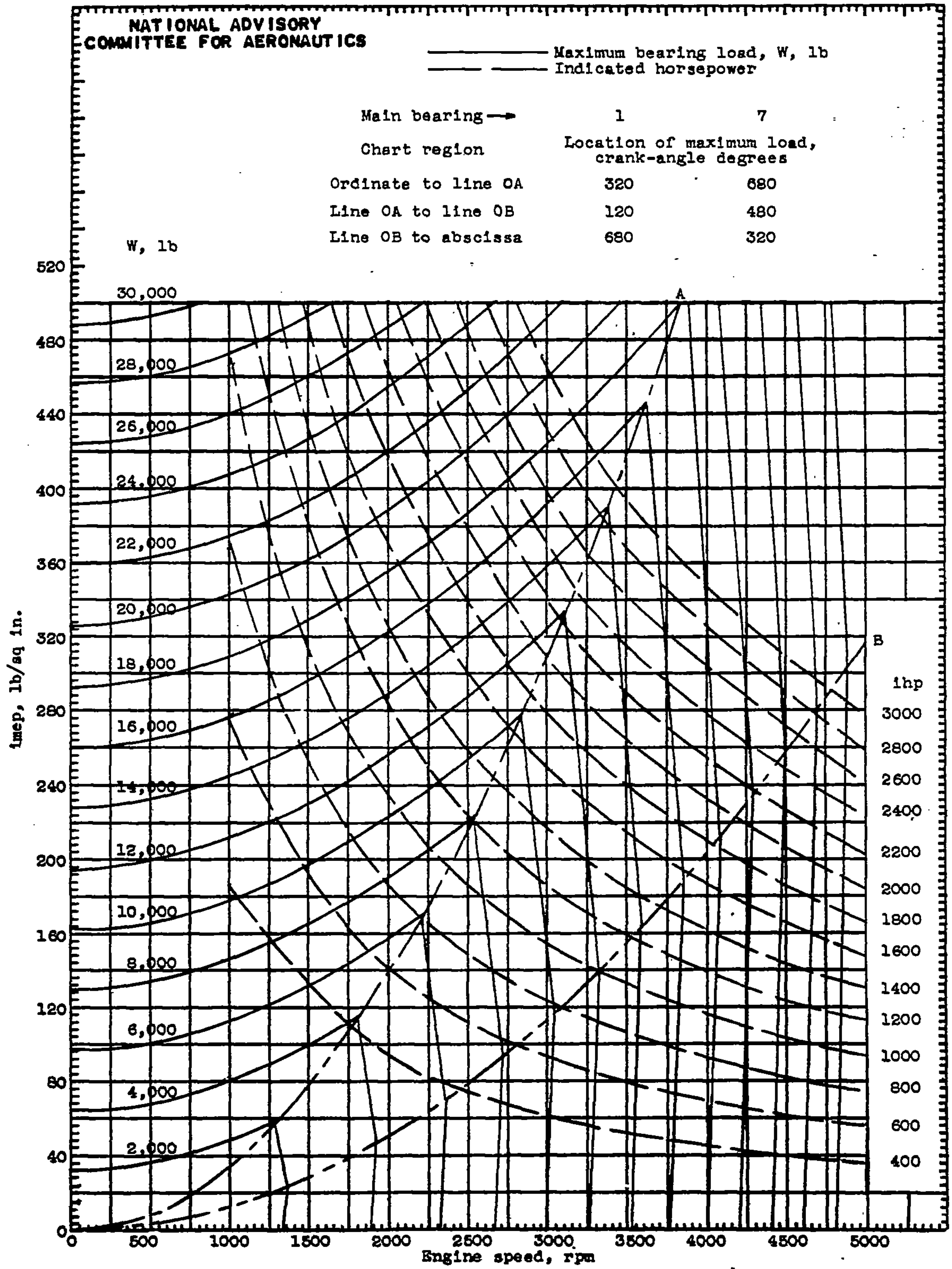

(c) Compression ratio, 8.50.
Figure 28. - Concluded.